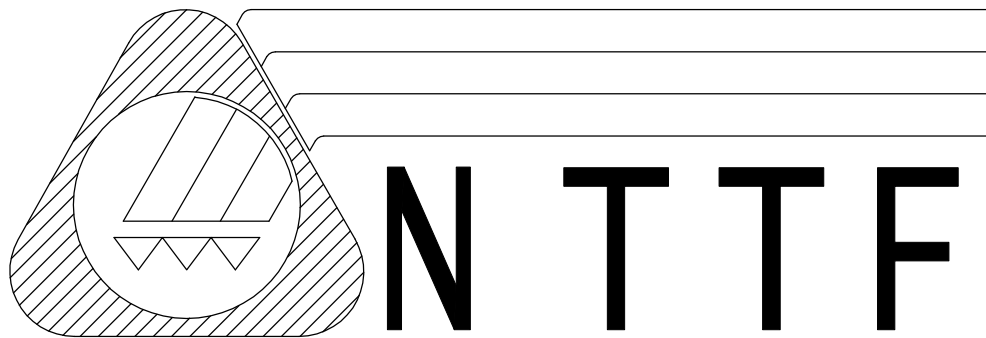


PRESS TOOL

STANDARDS



**COMPILED BY SRIVINAY
PGDTD 2009,
NTTF BANGALORE**

LIST OF GENERAL NOTES TO BE ADDED IN PRESS TOOLS
(INCORPORATE RELEVANT NOTES IN TOOL DRAWINGS)

MATERIAL

CONDITION/SURFACE TREATMENT

SPECIFICATION

EQUIVALENT SPECIFICATION

THICKNESS

STRIP WIDTH

TYPE OF FEED-ROLL/STRIP

SHEAR STRENGTH

CLEARANCE/SIDE

SHUT HEIGHT

STROKE

STRIPPER STROKE

KNOCKOUT STROKE

TONNAGE REQUIRED

STRIPPER FORCE

BLANK HOLDING FORCE

TOOL TO BE LOADED ON

COMPONENT DRAWING NO

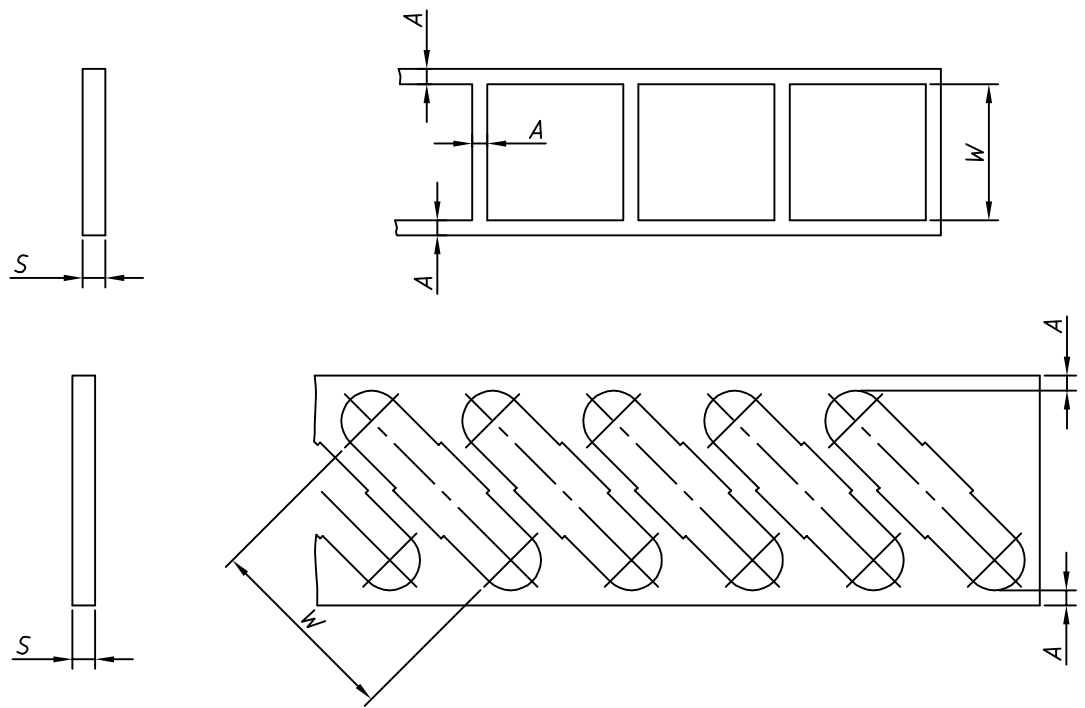
PREVIOUS OPERATION

SUBSEQUENT OPERATION

FOR SEQUENCE OF OPERATIONS,REFER NTTF DRG NO

BLANK DIMENSIONS TO BE FINALISED AFTER.....

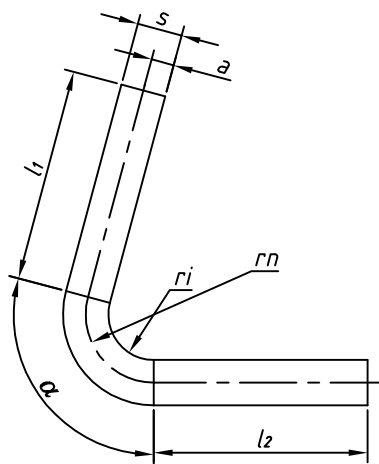
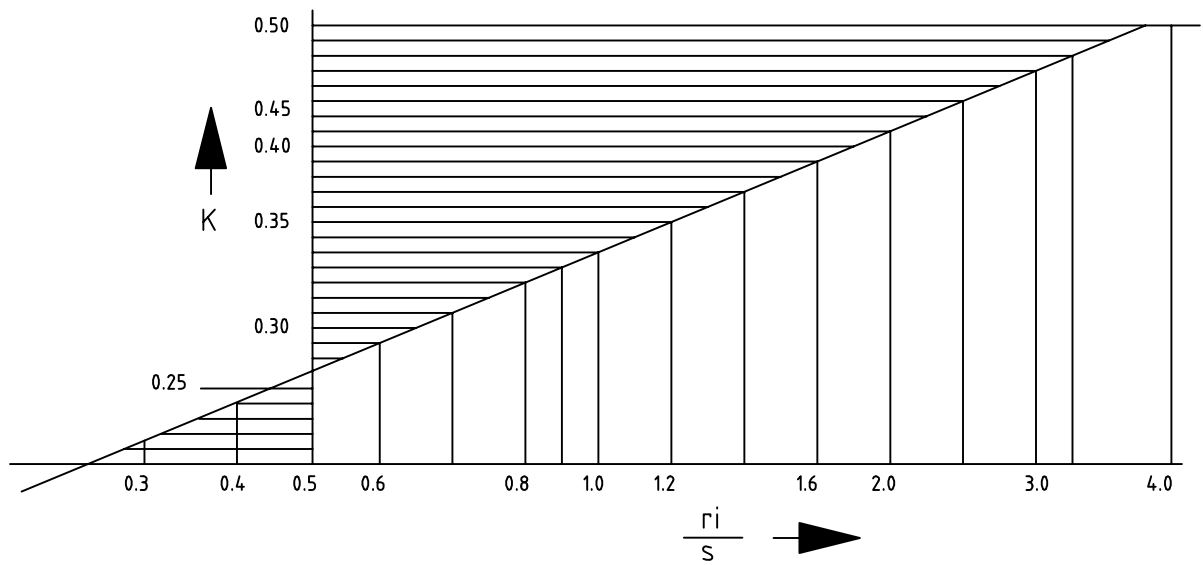
DRN	01/10/09	SHAMNA MOHAN	
CHD	-	-	
APPR.	-	-	
NTTF BANGALORE			05-00-01



1. THE STRIP WIDTH SHOULD BE ROUNDED OFF TO FULL mm. IN EXCEPTIONAL CASES eg; HIGH QUANTITY, COSTLY MATERIAL etc. CALCULATED STRIP WIDTH SHOULD BE ROUNDED OFF TO 0.5mm.
2. CONSIDER STRIP WIDTH OR LENGTH OF PART "W" WHEN PLACED AT AN INCLINATION WHICHEVER IS GREATER FOR CALCULATING SCRAP BRIDGE 'A'.
3. FOR HARD PAPER,BAKELITE & MOULDED SHEET, BRIDGE = 2A.

STOCK MATERIAL	S	W			
		UP TO 50mm	50 TO 100	100 TO 150	ABOVE 150
FERROUS METALS	UP TO 0.5	1.5	2.0	3.0	3.5
	0.5 TO 1.0	1.5	2.0	3.0	3.5
BRASS	1.0 TO 2.0	2.0	2.5	3.0	3.5
	2.0 TO 3.0	2.0	3.5	4.0	4.5
BRONZE	3.0 TO 4.0	2.5	4.0	4.5	5.0
	ABOVE 4.0	0.8x5	1x5	1.2x5	1.5x5
ALUMINIUM, ZINC & OTHER SOFTER METALS	UP TO 0.5	2.4	3.6	4.8	6.0
	0.5 TO 1.0	1.8	2.4	3.6	4.8
	1.0 TO 2.0	3.0	3.6	4.2	5.4
	2.0 TO 3.0	3.6	4.8	6.0	7.2
	3.0 TO 4.0	4.8	6.0	7.2	8.4
	ABOVE 4.0	1x5	1.2x5	1.9x5	2x5

DRN	01/10/09	SHAMNA MOHAN		
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			BACK SCRAP, FRONT- SCRAP & SCRAP BRIDGE	05-00-04



$$a = s \cdot K \text{ mm}$$

$$r_n = r_i + a \text{ mm}$$

$$l_0 = l_1 + l_2 + \frac{\pi r_n}{180^\circ} \times \alpha^\circ$$

$$\text{For } \frac{r_i}{s} \leq 0.5 \quad K = 0.27$$

DRN	01/10/09	SHAMNA MOHAN		
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			BEND ALLOWANCE	05-00-06

	CUTTING CLEARANCE AS % OF SHEET THICKNESS	
	UP TO 3mm THICK	ABOVE 3mm THICK
STEEL	5%	7%
BRASS	4%	5%
COPPER	3%	4%
ALUMINIUM	2%	3%

FOR STEEL SHEAR STRESS ABOVE 60kg/mm²

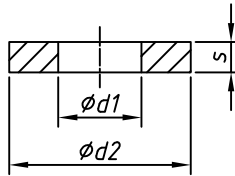
CUTTING CLEARANCE = $1.4C$

WHERE C=CUTTING CLEARANCE PER SIDE FROM ABOVE TABLE

DRN	01/10/09	SHAMNA MOHAN		
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			CUTTING CLEARANCE	05-00-07

MATERIAL	SHEAR STRENGTHkg/mm ²
STEEL-0.1% CARBON	24-30
STEEL-0.2% CARBON	32-40
STEEL-0.3% CARBON	36-48
STEEL-0.4% CARBON	45-56
STEEL-0.6% CARBON	55-70
STEEL-0.8% CARBON	70-90
SILICON STEEL	45-55
STAINLESS STEEL	30-40
COPPER	20-30
BRASS 60	35-40
BRONZE	36-45
TIN	3-4
ZINC	10-12
LEAD	2-3
ALUMINIUM (99.5%)	8-12
ALUMINIUM-ALLOY(WITH Mg AND Si)	15-23
PAPER OR CARD BOARD	2-5
HARD BOARD	7-9
PAPER(LAMINATED/RESIN IMPREGINATED)	10-14
FABRIC(LAMINATED/RESIN IMPREGINATED)	9-12
MICA	5-8
PLYWOOD	2-4
CELLULOID	4-6
LEATHER	0.7
SOFT RUBBER	0.7
HARD RUBBER	2-6

DRN	01/10/09	SHAMNA MOHAN	REFER:OEHLER(P-28)	
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			SHEAR STRENGTH	05-00-08

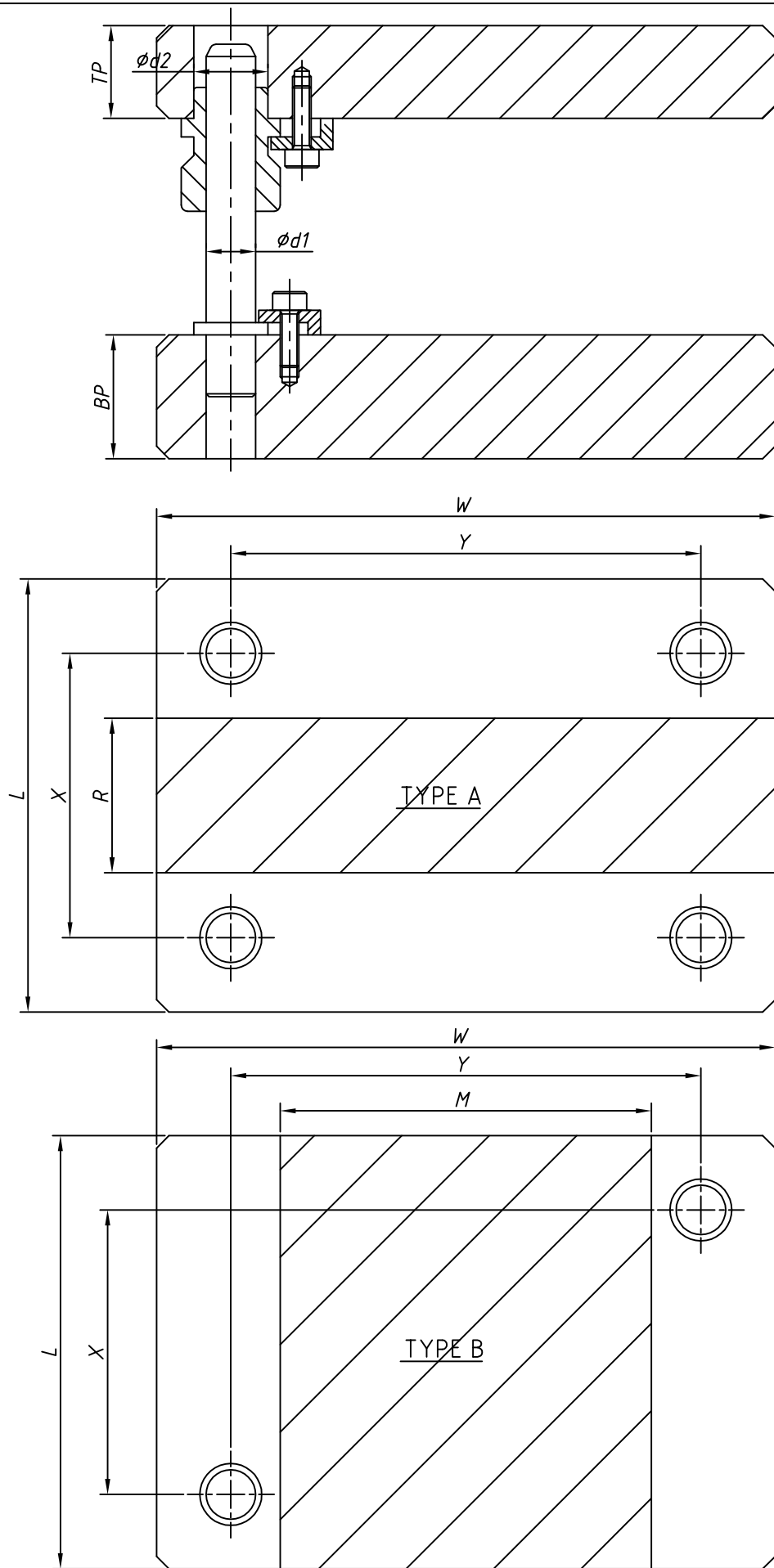


d1	d2	s
7	16	3
9	20	4
11	24	4
13	30	5
18	42	6

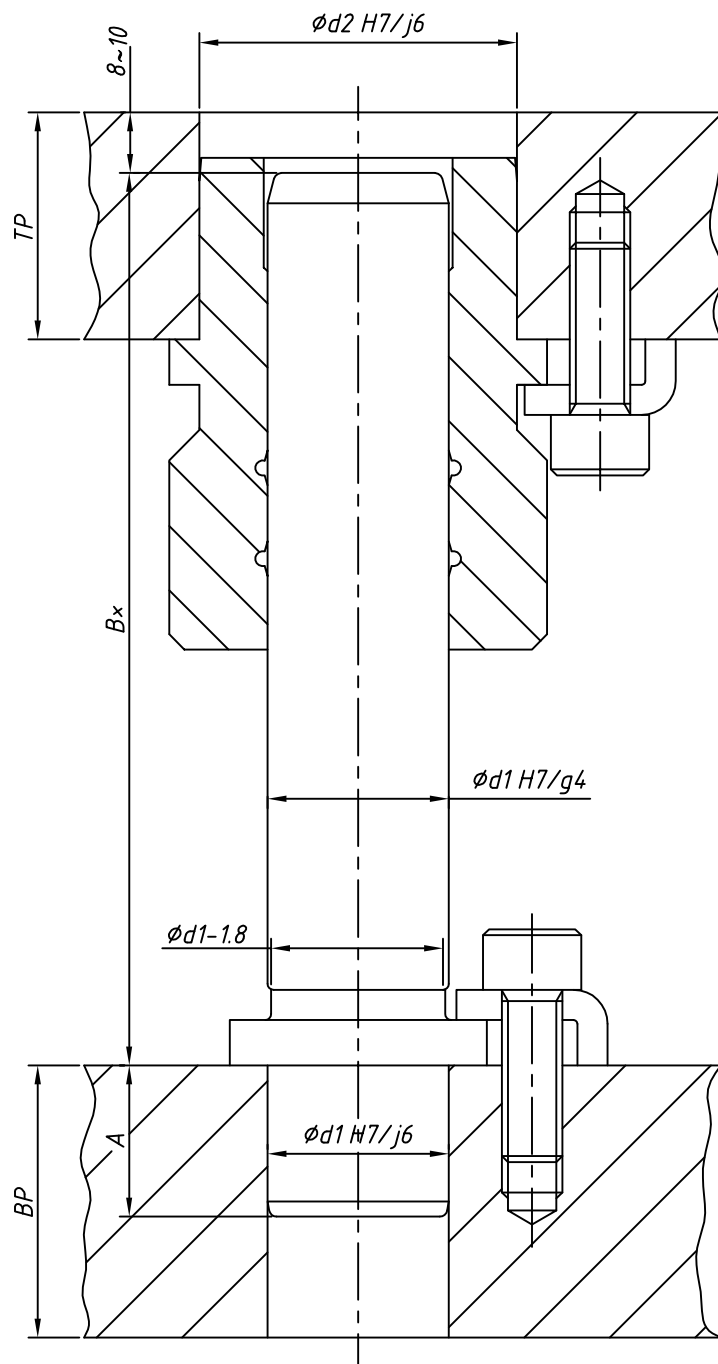
Washer with dia $d1=9$ should designed as
04-00-34/9

Gen.Tol. ± 0.2

DRN 01/10/09 SHAMNA MOHAN CHD - - APPR. - -			WASHER		K100			
			POS	DESCRIPTION	ITEM	MAT.	SIZE	LENGTH
			REMARKS					
NTTF BANGALORE			WASHER				04-00-34	



DRN	01/10/09	SHAMNA MOHAN		
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			DIE SET STANDARD SIZES	05-02-01-1,3-1



CHOOSE 'B' SUCH THAT
BUSH DOES NOT DISENGAGE
FROM PILLAR AT MAXIMUM
STROKE.

d1	H7		g4		H7/g4		j6		H7/j6	
	e1	es	e1	es	MIN CL	MAX CL	e1	es	MAX INT	MAX CL
24/25	0	+21	-13	-7	7	34	-4	+9	9	25
30/32	0	+25	-16	-9	9	41	-5	+11	11	30
40/42	0	+25	-16	-9	9	41	-5	+11	11	30
50/52	0	+30	-18	-10	10	48	-7	+12	12	37

NOTE:
INT-INTERFERENCE
CL-CLEARANCE

DRN	01/10/09	SHAMNA MOHAN	
CHD	-	-	
APPR.	-	-	
NTTF BANGALORE			DEMOUNTABLE PILLAR AND BUSH ASSEMBLY
			05-02-01-1,3-2

DRN	01/10/09	SHAMNA MOHAN
CHD	-	-
APPR.	-	-
NTTF BANGALORE		

DIE SET STANDARD SIZES		05-02-01,1,3-3
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SERIES	L	W	X	Y	TP	BP	TYPE A			TYPE B			ød1	ød2
							R	W		M	L			
1515	146	146	80	80	26/36	26/36	37	146		37	146		18/19	35
1520	146	196	80	130	26/36	26/36	37	196		87	146		18/19	35
2020	196	196	120	120	36/46/56	36/46/56	50	196		50	196		24/25	42
2025	196	246	100	150	36/46/56	46/56/66	40	246		90	196		30/32	50
2030	196	296	100	200	36/46/56	46/56/66	40	296		140	196		30/32	50
2035	196	346	100	250	46/56/66	56/66/76	40	346		190	196		30/32	50
2525	246	246	150	150	26/36/46/56	36/46/56/66	90	246		90	246		30/32	50
2530	246	296	150	200	36/46/56	46/56/66	90	296		140	246		30/32	50
2535	246	346	150	250	36/46/56/66	46/56/66	90	346		190	246		30/32	50
3030	296	296	200	200	46/56/66	46/56/66	140	296		140	296		30/32	50
3035	296	346	200	250	46/56/66	46/56/66	140	346		190	296		30/32	50
3035×	296	346	172	222	46/56/66	46/56/66	91	346		141	296		50/52	72
3040	296	396	186	286	56/66/76	56/66/76	117	396		217	296		40/42	60
3040×	296	396	172	272	56/66/76	56/66/76	91	396		191	296		50/52	72
3045	296	446	186	336	56/66/76	56/66/76	117	446		267	296		40/42	60
3045×	296	446	172	322	56/66/76	56/66/76	91	446		241	296		50/52	72
3050	296	496	172	372	56/66/76	56/66/76	91	496		291	296		50/52	72

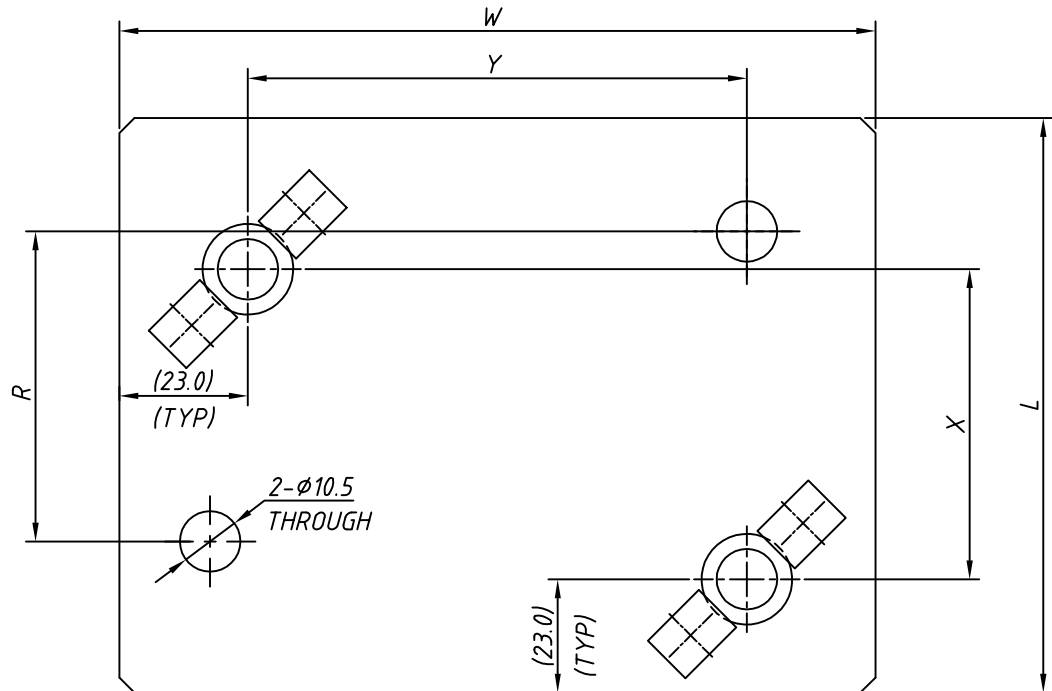
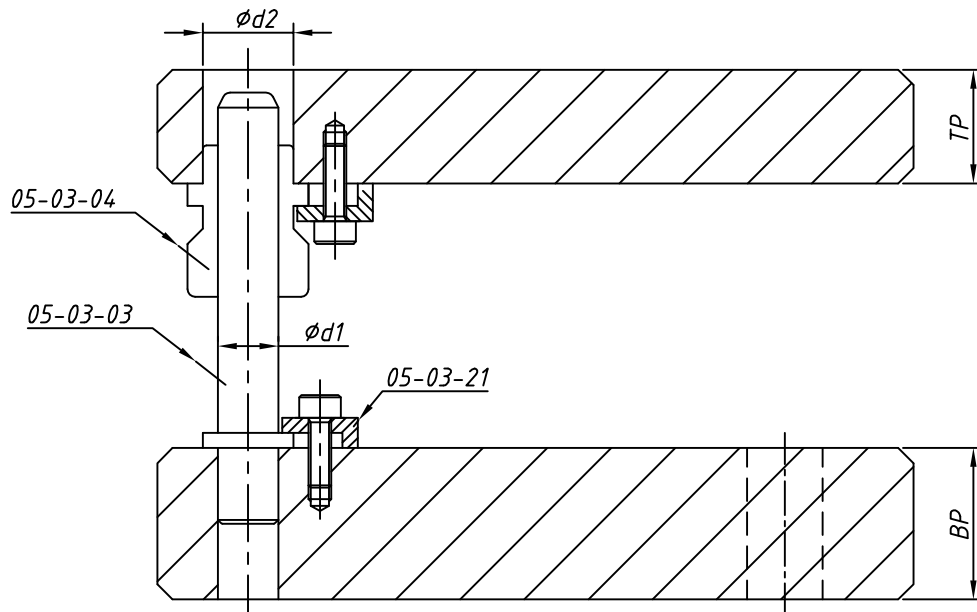
1. ×FOR PRESS STROKE=160mm
2. R×W AND M×L FOR 3CLAMP ARRANGEMENT REFER 05-03-23-4,0-4
3. FOR LARGER R×W AND M×L USE 2CLAMP ARRANGEMENT REFER 05-03-23-4,0-3
4. SELECTION OF DIE SET STANDARDS FOR DIFFERENT M/C S REFER 05-02-02-1&2

DRN	01/10/09	SHAMNA MOHAN
CHD	-	-
APPR.	-	-
NTTF BANGALORE		

DIE SET STANDARD SIZES		05-02-01,1,3-4
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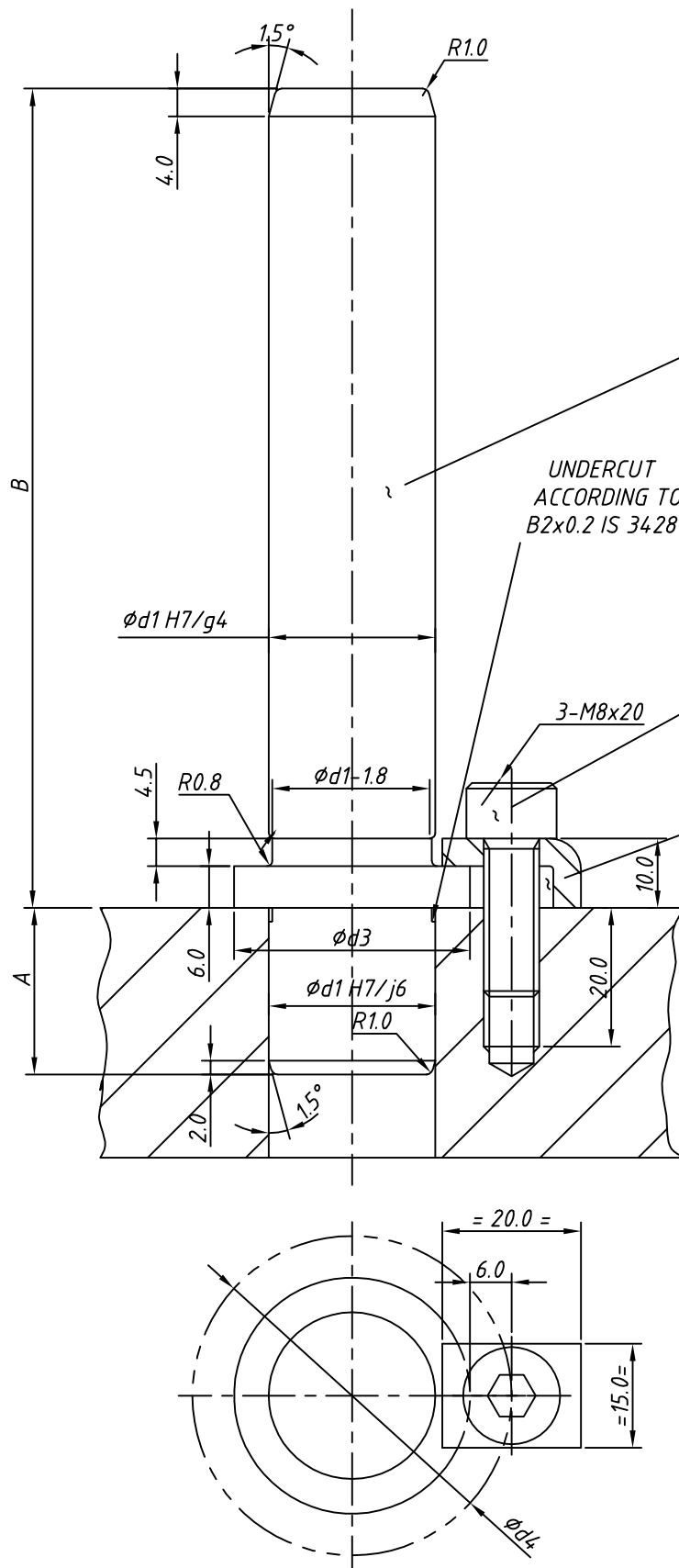
SERIES	L	W	X	Y	TP	BP	TYPE A			TYPE B			ød1	ød2
							R	W		M	L			
3535	346	346	236	236	46/56/66	46/56/66	167	346		167	346		40/42	60
3535×	346	346	222	222	46/56/66	46/56/66	141	346		141	346		50/52	72
3540	346	396	236	286	56/66/76	56/66/76	167	396		217	346		40/42	60
3540×	346	396	222	272	56/66/76	56/66/76	141	396		191	346		50/52	72
3545	346	446	236	336	56/66/76	56/66/76	167	446		267	346		40/42	60
3545×	346	446	222	322	56/66/76	56/66/76	141	446		241	346		50/52	72
3550	346	496	222	372	56/66/76	56/66/76	141	496		291	346		50/52	72
4040	396	396	286	286	56/66/76	56/66/76	217	396		217	396		40/42	60
4040×	396	396	272	272	56/66/76	56/66/76	191	396		191	396		50/52	72
4045	396	446	272	322	56/66/76	56/66/76	191	446		241	396		50/52	72
4050	396	496	272	372	56/66/76	56/66/76	191	496		291	396		50/52	72
4060	396	596	272	472	56/66/76	56/66/76	191	596		391	396		50/52	72
4540×	446	396	322	272	56/66/76	56/66/76	241	396		191	446		50/52	72
4545	446	446	322	322	56/66/76	56/66/76	241	446		241	446		50/52	72
4550	446	496	322	372	56/66/76	56/66/76	241	496		291	446		50/52	72
4560	446	596	322	472	56/66/76	56/66/76	241	596		391	446		50/52	72

1. xFOR PRESS STROKE=160mm
2. RxW AND MxL FOR 3CLAMP ARRANGEMENT REFER 05-03-23-4,0-4
3. FOR LARGER RxW AND MxL USE 2CLAMP ARRANGEMENT REFER 05-03-23-4,0-3
4. SELECTION OF DIE SET STANDARDS FOR DIFFERENT M/C S REFER 05-02-02-1&2 AND 05-02-02-02-2



	BP	TP	L	W	X	Y	R	M	$\phi d1$	$\phi d2$
AJ&JFP	26/36	26	100	160	54	114	70	110	18/19	$\phi 26$
DP	26/36	26/36	146	246	100	200	104	170	18/19	$\phi 26$

DRN	01/10/09	SHAMNA MOHAN			
CHD	-	-			
APPR.	-	-			
NTTF BANGALORE			TOGGLE PRESS DIE SET		
			05-02-03		



TO SPECIFY WRITE

05-03-02/ $\phi d1$ /B

01

UNDERCUT
ACCORDING TO
B2x0.2 IS 3428

02

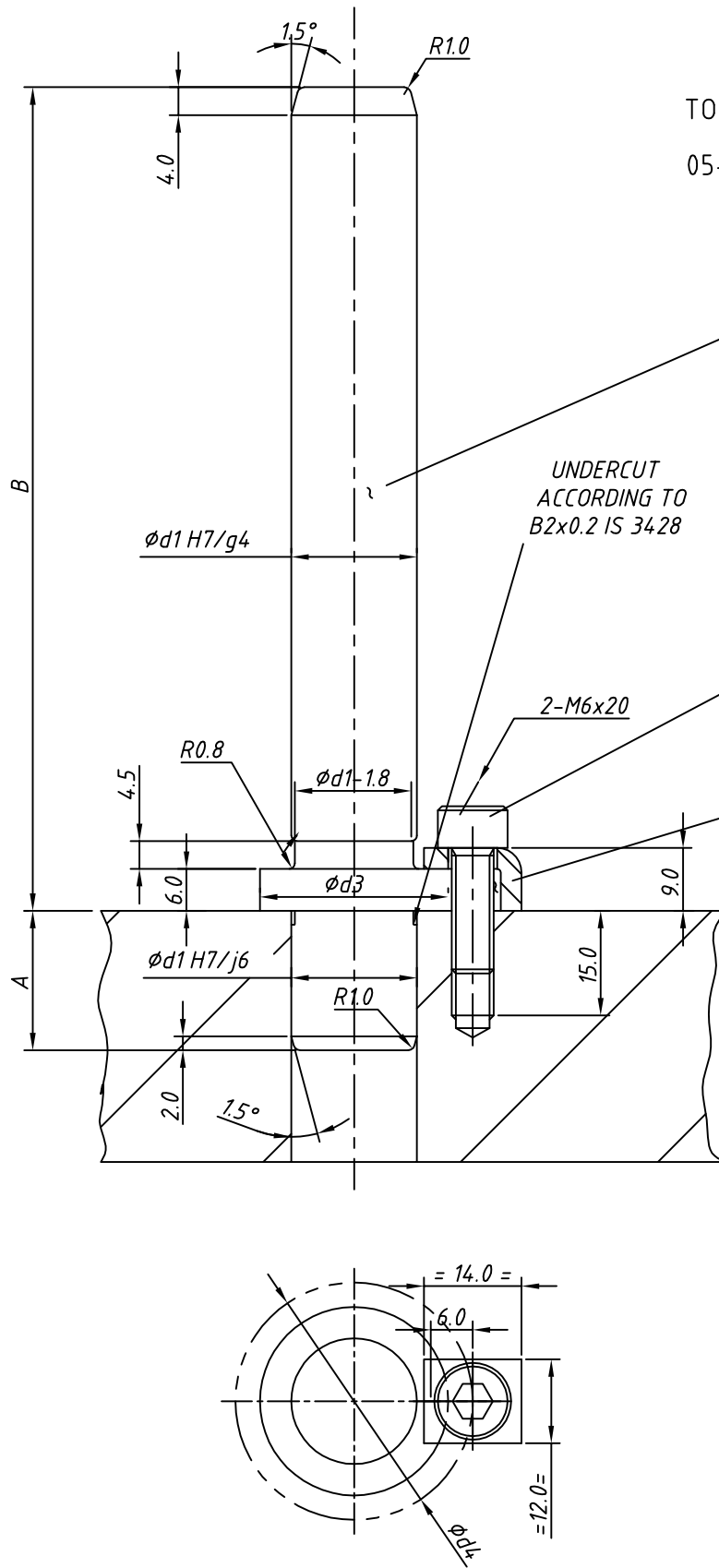
03

*

$\phi d1$	$\phi d3$	$\phi d4$	A
24/25	34	46	24
30/32	40	52	30
40/42	50	62	40
50/52	60	72	50

× REFER -05-03-22

DRN	01/10/09	SHAMNA MOHAN	3	CLAMP		MS			
CHD	-	-	2	SHCS			M8	25	
APPR.	-	-	1	GUIDE PILLAR		E230	05-03-02		CHDN
POS.	DESCRIPTION	QTY.	MAT.	FINISHED SIZE	LENGTH	REMARKS			
NTTF BANGALORE			DEMOUNTABLE GUIDE PILLAR ASSEMBLY				05-03-02		



TO SPECIFY WRITE

05-03-03/ $\phi d1$ /B

01

UNDERCUT
ACCORDING TO
B2x0.2 IS 3428

02

03

*

$\phi d1$	$\phi d3$	$\phi d4$	A
18/19	27	34	20

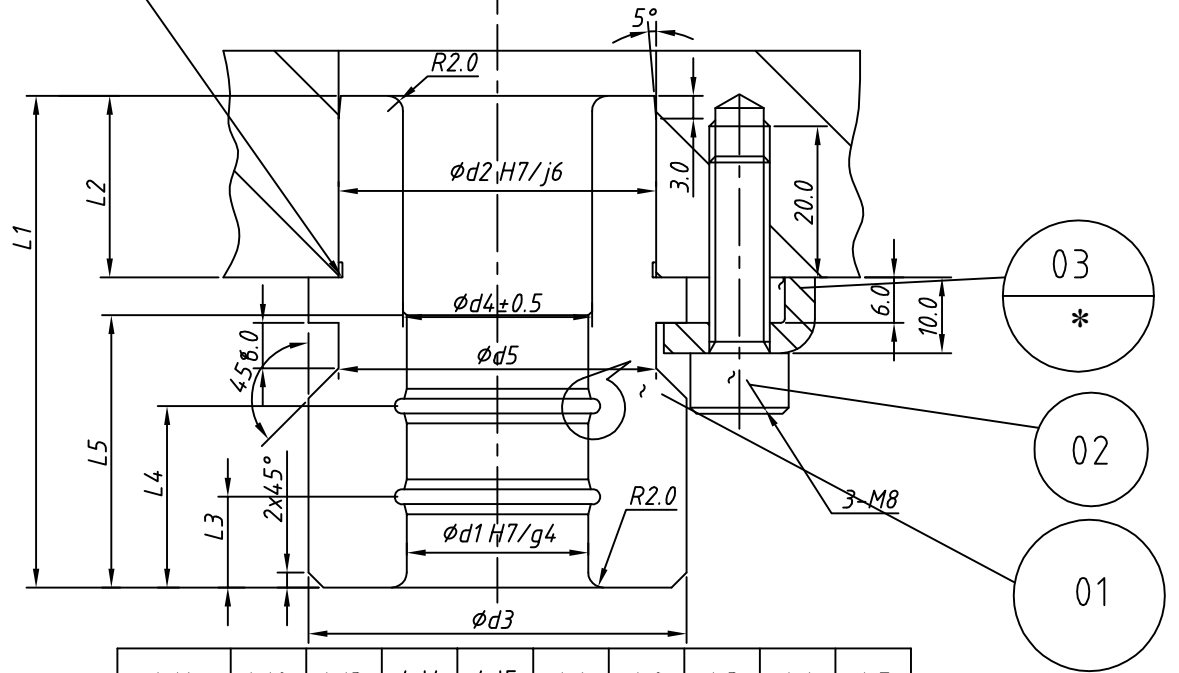
× REFER -05-03-21

DRN	01/10/09	SHAMNA MOHAN	3	CLAMP		MS			
CHD	-	-	2	SHCS		M6	20		
APPR.	-	-	1	GUIDE PILLAR		E230	05-03-02		CHDN
			POS.	DESCRIPTION	QTY.	MAT.	FINISHED SIZE	LENGTH	REMARKS
NTTF BANGALORE			DEMOUNTABLE GUIDE PILLAR ($\phi 18/19$)				05-03-03		

05-03-16/φd1/L1

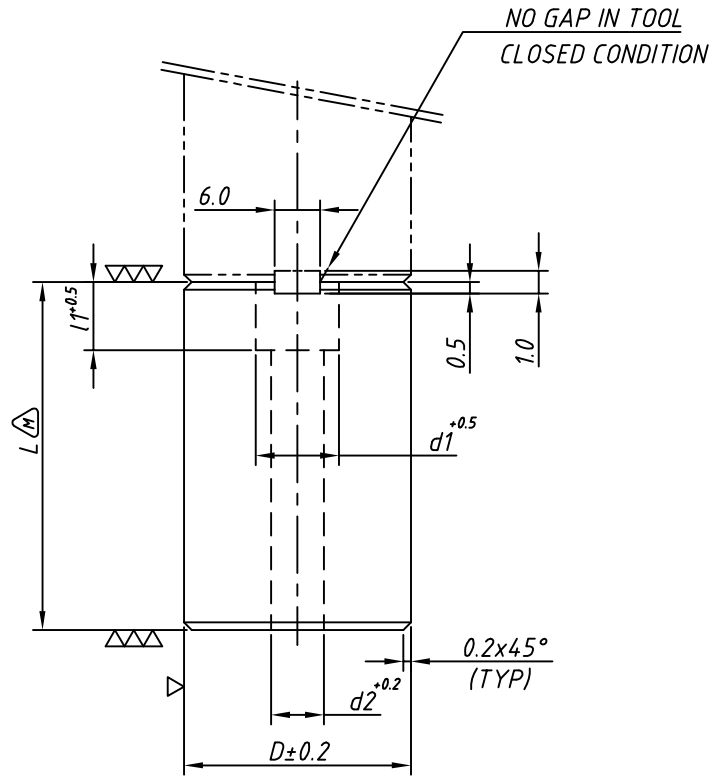
× REFER -05-03-21

UNDERCUT
ACCORDING TO
B2x0.2 IS 3428



φd1	φd2	φd3	φd4	φd5	L1	L2	L3	L4	L5
24/25	42	50	64	42	90	24	12	24	36
30/32	50	58	72	50	90	30	18	36	45
30/32	50	58	72	50	110	30	18	36	45
40/42	60	68	82	60	90	40	25	50	60
40/42	60	68	82	60	110	40	25	50	60
40/42	60	68	82	60	125	40	25	50	60
50/52	72	80	94	72	90	50	28	56	75
50/52	72	80	94	72	110	50	28	56	75
50/52	72	80	94	72	125	50	28	56	75
50/52	72	80	94	72	145	50	28	56	75
50/52	72	80	94	72	166	50	28	56	75

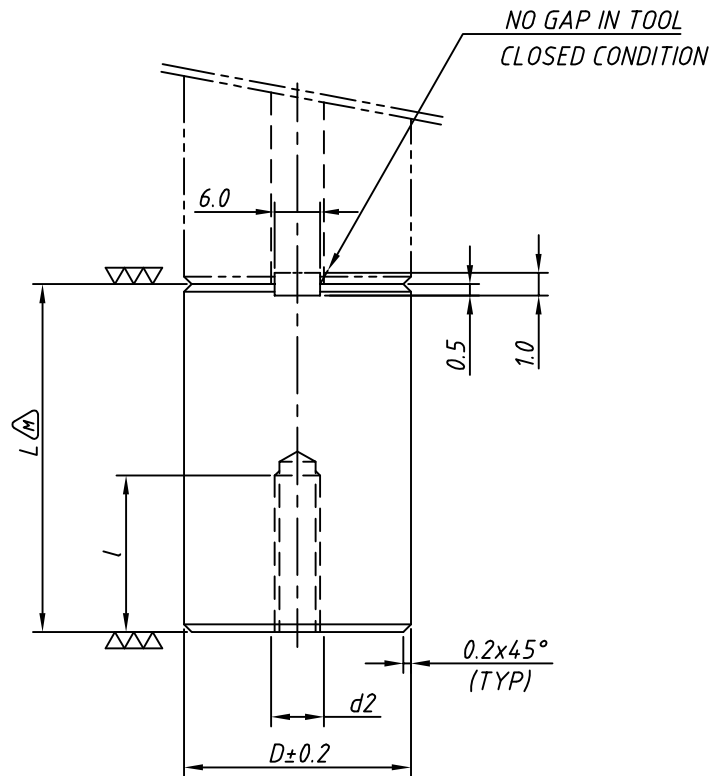
				3	CLAMP	3	MS		
DRN	01/10/09	SHAMNA MOHAN		2	SHCS	3		M8	25
CHD	-	-		1	GUIDE BUSH		E230		CHDN
APPR.	-	-		POS.	DESCRIPTION	QTY.	MAT.	FINISHED SIZE	LENGTH
NTTF BANGALORE				DEMOUNTABLE GUIDE BUSH				05-03-16	



D	d1	d2	l1	L
20-30	11.0	7.0	9.0	
30-45	14.0	9.0	11.0	
ABOVE 45	17.0	11.0	13.0	

TO SPECIFY WRITE 05-03-18/D/L

DRN	01/10/09	SHAMNA MOHAN	MATERIAL:K720	HRC:55-58
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			SETTING BOLCK TYPE 1	05-03-18

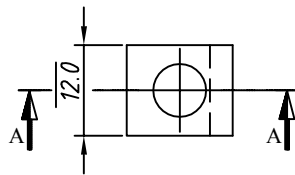
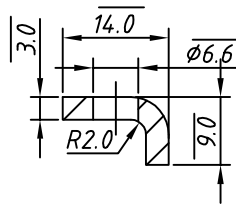


D	d	l	L
15-25	M6	15	
25-45	M8	20	
ABOVE 45	M10	25	

TO SPECIFY WRITE 05-03-19/D/L

DRN	01/10/09	SHAMNA MOHAN	MATERIAL:K720	HRC:55-58
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			SETTING BOLCK TYPE 2	05-03-19

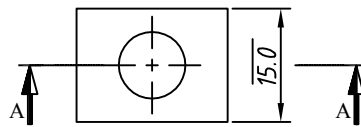
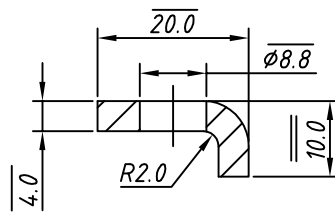
CLAMP FOR DIE SET SERIES 1515 AND 1520



TO SPECIFY WRITE

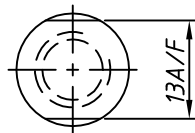
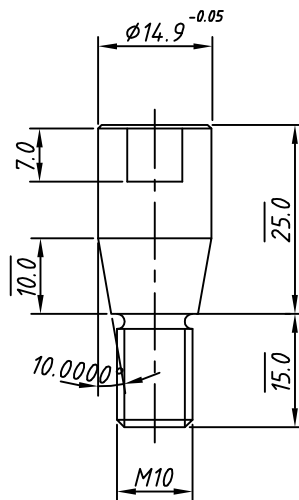
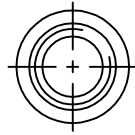
05-03-21

DRN	01/10/09	SHAMNA MOHAN	MATERIAL:MS	
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			CLAMP FOR $\phi 18/19$ PILLAR/BUSH	05-03-21



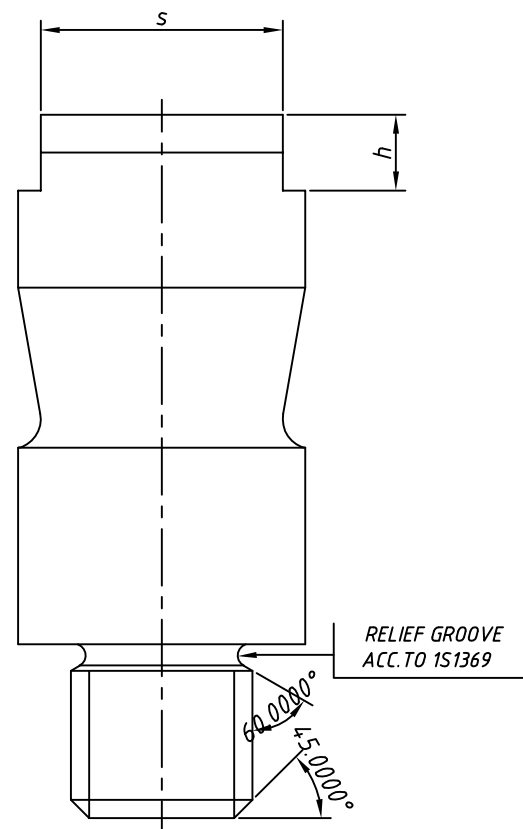
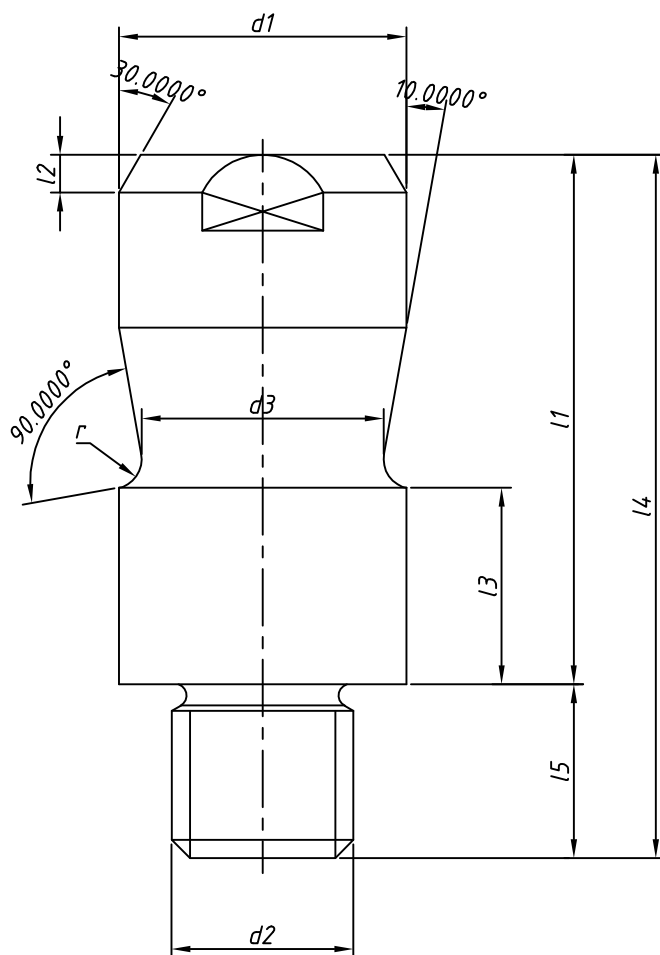
TO SPECIFY WRITE
05-03-22

DRN	01/10/09	SHAMNA MOHAN	MATERIAL:MS	
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			CLAMP	05-03-22



TO SPECIFY WRITE
05-03-24

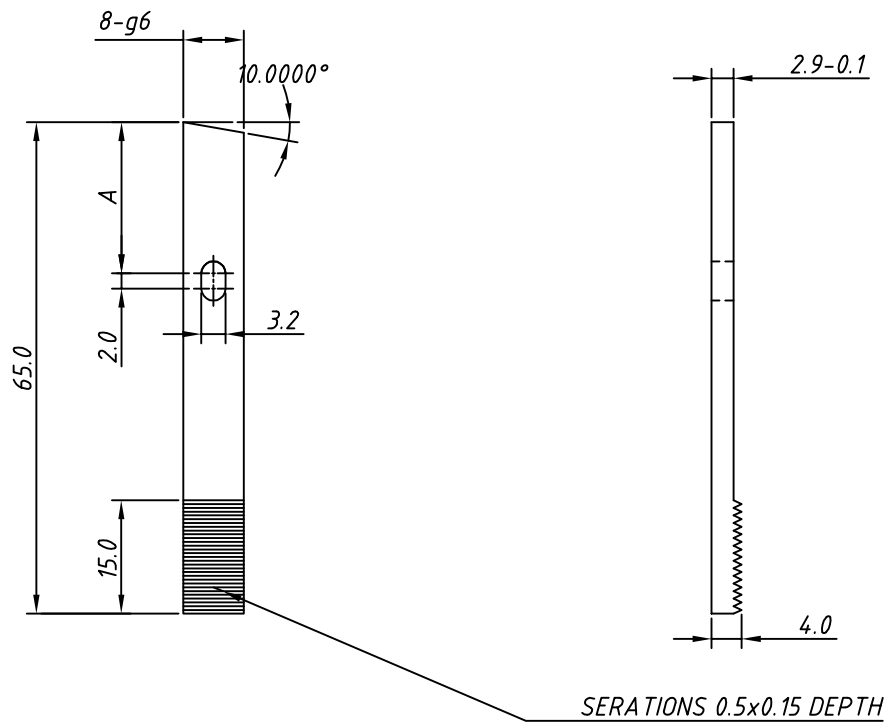
DRN	01/10/09	SHAMNA MOHAN	MATERIAL:MS	
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			SHANK FOR GENENCO TOGGLE PRESS	05-03-24



INDEX	d1	d2	d3	l1	l2	l3	l4	l5	h	r	s ^{h13}	PRESSES
1	16	M10x1.25	12	32	2.5	10	47	15	6	2	13	NTTF HAND TOGGLE
2	20	M16x1.5	15	40	3	12	58	18	6	2.5	17	
3	25	M16x1.5 M20x1.5	20	45	4	16	68	23	6	2.5	19	
4	32	M20x1.5 M24x1.5	25	56	4	16	79	23	8	2.5	27	BATLIBOI, GODREJ-100KN
5	38	M24x1.5	32	60	5	26	83	23	10	4	32	BEUTLER 80KN SARASWATHI 100KN
6	40	M30x1.5		70			93					GODREJ, AMETEEP-160, 250KN BATLIBOI-300KN, LAMBA-300KN
7	50	M30x2	42	80	6	26	108	28	12	4	41	GODREJ AMETEEP-400, 630KN GODREJ, HMT-100KN
8		M36x2										
9	63	M42x3	53	100	8	26	128	28	16	4	55	AMETEEP-800, 1000KN
10	80	M36x2 M42x3	-	70	10	-	106	36	20	-	65	HMT 1600KN

TO ORDER SHANK OF dia=40 WRITE 05-03-25/6(6 DENOTES INDEX NO.)

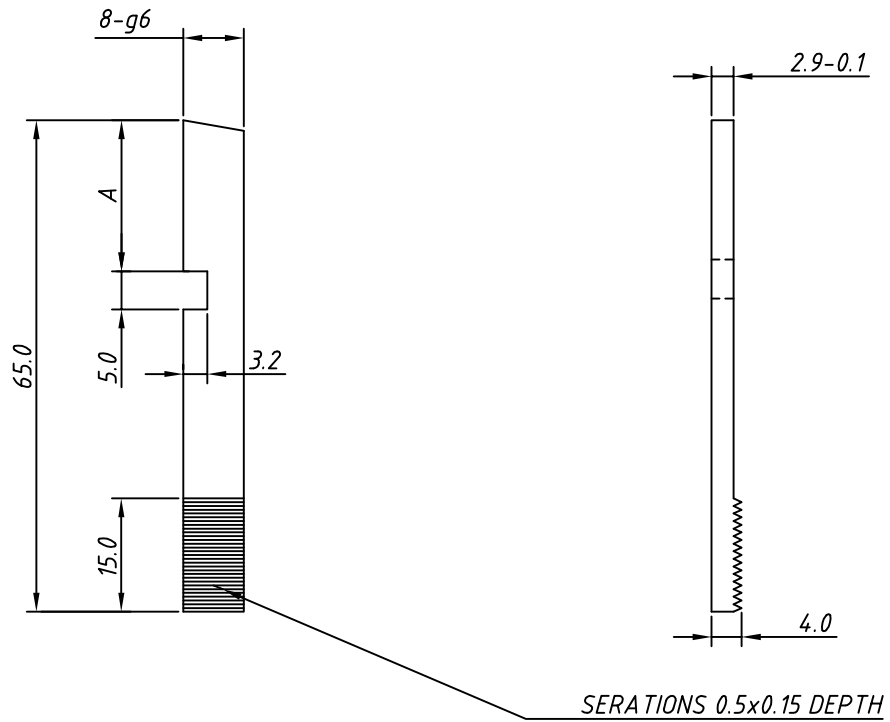
DRN	01/10/09	SHAMNA MOHAN	MATERIAL:MS	
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			SHANK	05-03-25



TO SPECIFY WRITE

05-03-26/A

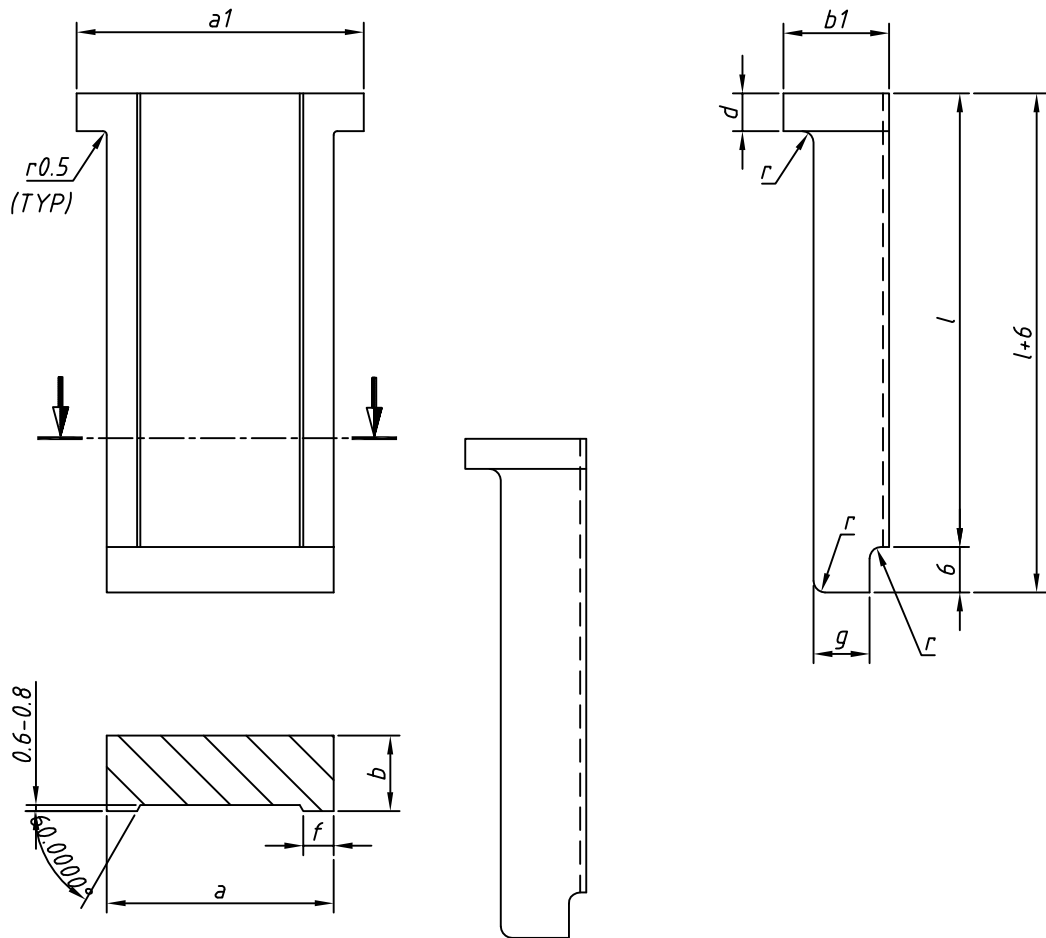
DRN	01/10/09	SHAMNA MOHAN	MATERIAL:K720 HRC:55-58	
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			FINGER STOP TYPE 1	05-03-26



TO SPECIFY WRITE

05-03-27/A

DRN	01/10/09	SHAMNA MOHAN	MATERIAL:K720 HRC:55-58	
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			FINGER STOP TYPE 2	05-03-27

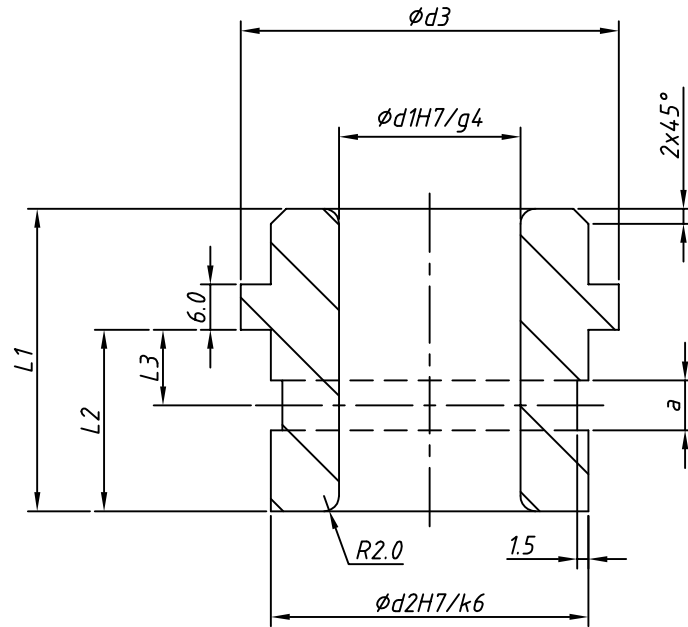


a=PITCH+0.05mm		a1	b	b1	d	f	g	r
Till - 6		a+6	6.5	9	4	1.2	4	1
6	10	a+6	6.5	9	4	1.6	4	1
10	16	a+6	6.5	9	4	2.5	4	1
16	25	a+7	8	11.5	5	3.0	5.5	1.5
25	40	a+8	10	14.0	5	4.0	7.4	1.5
40	100	a+10	12	16.0	5	5.0	9.0	1.5

TO SPECIFY WRITE

05-03-28/a/l

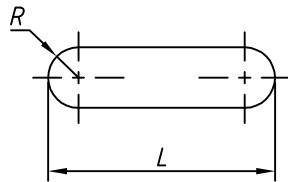
DRN	01/10/09	SHAMNA MOHAN	HARDNESS:HRC61 Tip hard&head HRC55 MAT.:K100	
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			PITCH PUNCH	05-03-28



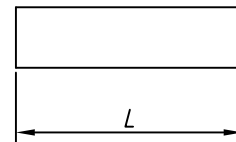
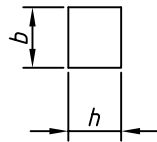
d1	d2	d3	a
24/25	42	50	6.5
30/32	50	58	8.5
40/42	60	68	8.5
50/52	72	80	8.5

TO SPECIFY WRITE
05-03-29/ $\phi d1/L1/L2/L3$

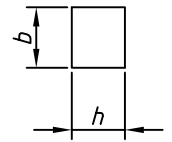
DRN	01/10/09	SHAMNA MOHAN	HARDNESS:CASE HARDENED	
CHD	-	-		
APPR.	-	-		
			MAT.:E230	
NTTF BANGALORE			STRIPPER PLATE GUIDE BUSH	05-03-29



TYPE A



TYPE B



bXh ^x	RANGE OF KEY LENGTH 'L'	
	MIN.	MAX.
2x2	6	20
3x3	6	36
4x4	8	45
5x5	10	56
6x6	14	70
8x7	18	90
10x8	22	110
12x8	28	140
14x9	36	160
16x10	45	180
18x11	50	200
20x12	56	220

PREFERRED LENGTH 'L'									
					6		8		10
	12		14		16		18		20
	22			25			28		
	32				36				40
				45					50
					56				
		63							70
									80
									90
									100

TO SPECIFY KEY OF b=10;h=8;L=40;TYPE-A
WRITE 05-03-31-10x8/40A

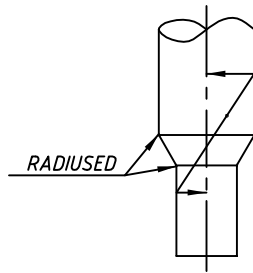
x

TOLERANCE ON 'b' IS 'h9'

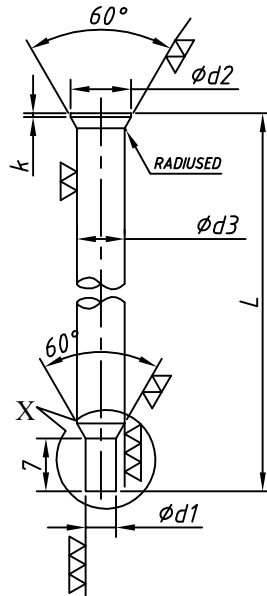
ON 'b' IS 'h9' FOR SQUARE SECTION

'h11' FOR RECTANGULAR SECTION

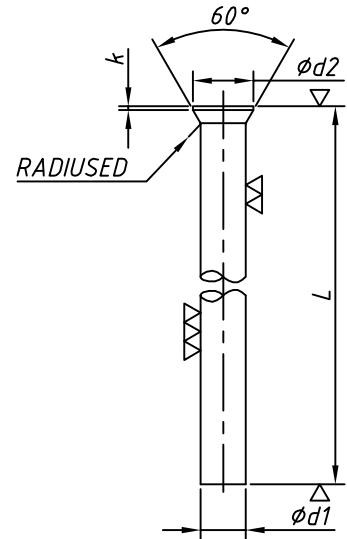
DRN	01/10/09	SHAMNA MOHAN	REMARK:ACC.TO I.S:2048 MAT.:K720	
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			PARALLEL KEY	05-03-32



DETAIL 'X'



TYPE C
WITH SHOULDRED HEAD



TYPE D
WITH PLAIN HEAD

NOMINAL SIZE d1-h6	d2		d3	k	LENGTH-L				
	TYPE-C	TYPE-D			TYPE-C		TYPE-D		
0.50	3	0.9	2	0.2	35	70	35	-	
0.55		1.0							
0.60		1.1							
0.65		1.2							
0.70&0.75	3	1.3	2	0.2	35	70	35	70	
0.80&0.85		1.4							
0.90&0.95		1.6							
1.00to1.10	3	1.8	2	0.5	35	70	35	70	
1.15to1.30		2.0							
1.35to1.50		2.2							
1.55to1.70	4.5	2.5	3	0.5	60	70	60	70	
1.75to1.90		2.8							
1.95to2.00		3.0							
2.05to2.20	4.5	3.2	3	0.5	60	70	60	70	
2.25to2.50		3.5							
2.55to2.95		4.0							
3.0to3.4	-	4.5	-	0.5	-	-	60	70	
3.5to3.9		5.0							
4.0to4.4		5.5							
4.5to4.9	-	6.0	-	0.5	-	-	60	70	
5.0to5.4		6.5							
5.5to5.9		7.0							
6.0to6.4	-	8.0	-	1.0	-	-	60	70	
6.5to7.4		9.0							
7.5to8.4		10							
8.5to9.4	-	11	-	1.0	-	-	60	70	
9.5to10.4		12							
10.5to11.4		13							
11.5to12.4	-	14	-	1.0	-	-	60	70	
12.5to13.4		15							
13.5to14.4		16							

MATERIAL : HSS

MATERIAL : SHANK HRC 61 ±2

EXECUTION : HARDENED&TEMPERED

HEAD HRC 45 ±6

DIA.d1 : FROM 0.5 TO 2.95 IN STEPS OF 0.5 AND FROM 3.0 TO14.4 IN STEPS OF 0.1

TO SPECIFY A ROUND PIERCING PUNCH ,TYPE-C,d1=1.9 AND L=60

WRITE : 05-03-33-C 1.9x60

DRN	01/10/09	SHAMNA MOHAN	REMARK:ACC.TO I.S:4296	
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			PARALLEL KEY	05-03-33

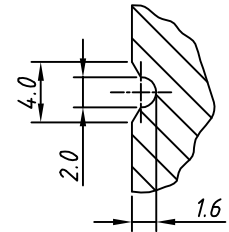
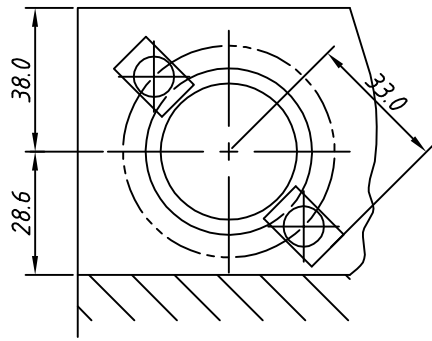
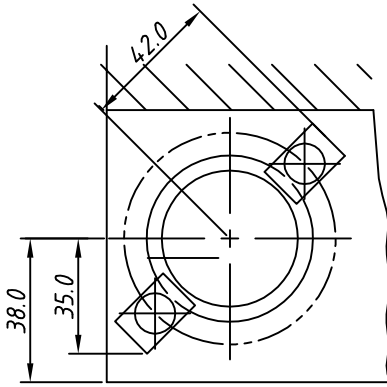


FIG 1

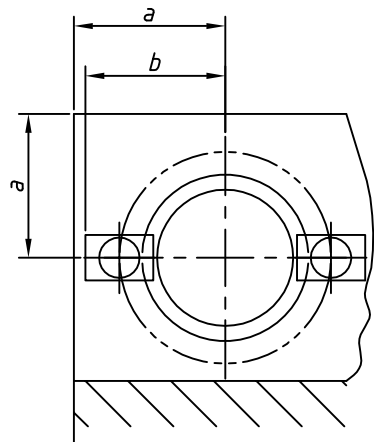
NTTF BANGALORE	DEMOUNTABLE BALL GUIDE BUSH(Ø18/19)	05-03-04
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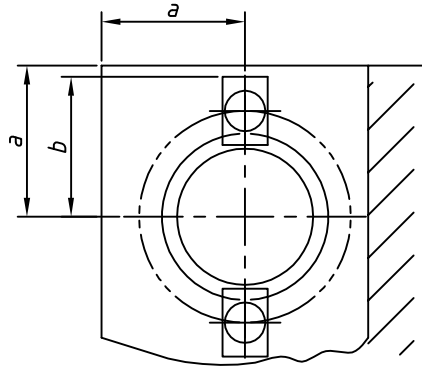
SERIES 2020
CLAMP DETAILS OF PILLAR
TYPE 'A' & 'B' DIE SET



SERIES 2020
CLAMP DETAILS OF BUSH
TYPE 'A' & 'B' DIE SET



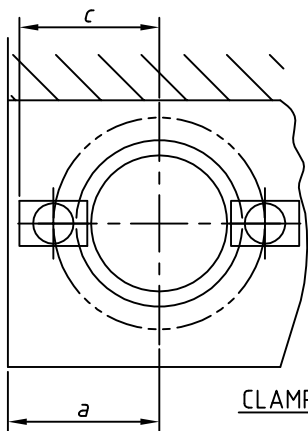
TYPE 'A' DIE SET



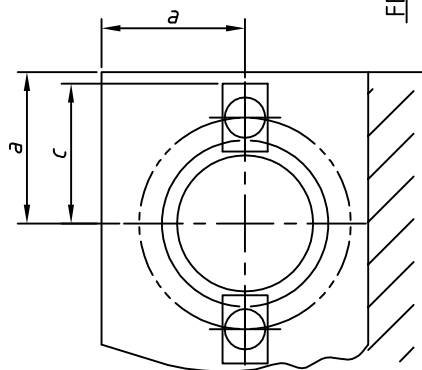
TYPE 'B' DIE SET

FEED

FEED



TYPE 'A' DIE SET



TYPE 'B' DIE SET

CLAMP DETAILS OF PILLAR BUSH

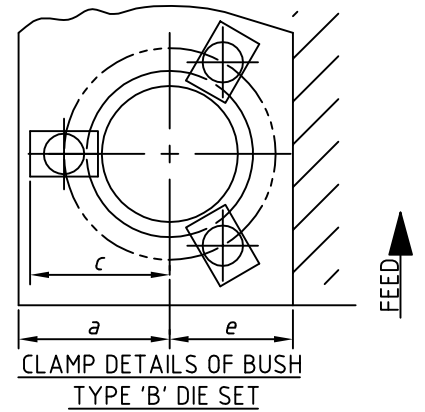
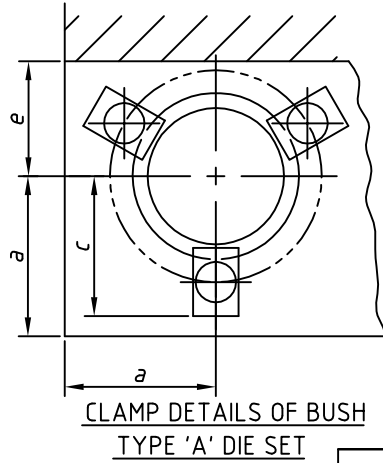
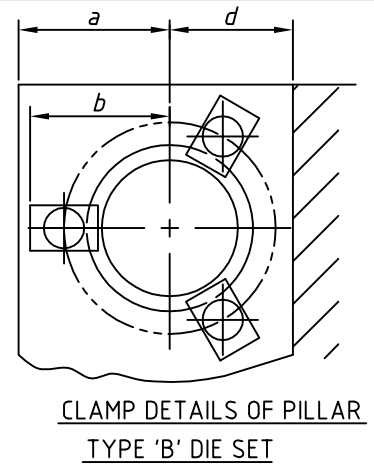
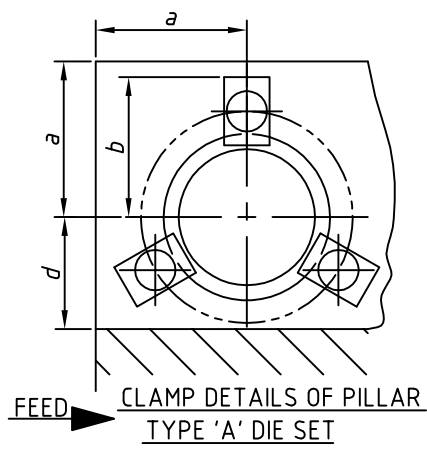
SERIES	a	b	c	Ød1
2025	48	36	46	30/32
2030	48	36	46	30/32
2035	48	36	46	30/32
3030	48	36	46	30/32
3035	48	36	46	30/32
3035	62	46	57	50/52
3040	55	41	51	40/42
3040	62	46	57	50/52
3045	55	41	51	40/42
3045	62	46	57	50/52
3050	62	46	57	50/52
4040	55	41	51	40/42
4040	62	46	57	50/52
4045	62	46	57	50/52
4050	62	46	57	50/52
4060	62	46	57	50/52

DRN	01/10/09	SHAMNA MOHAN
CHD	-	-
APPR.	-	-

NTTF
BANGALORE

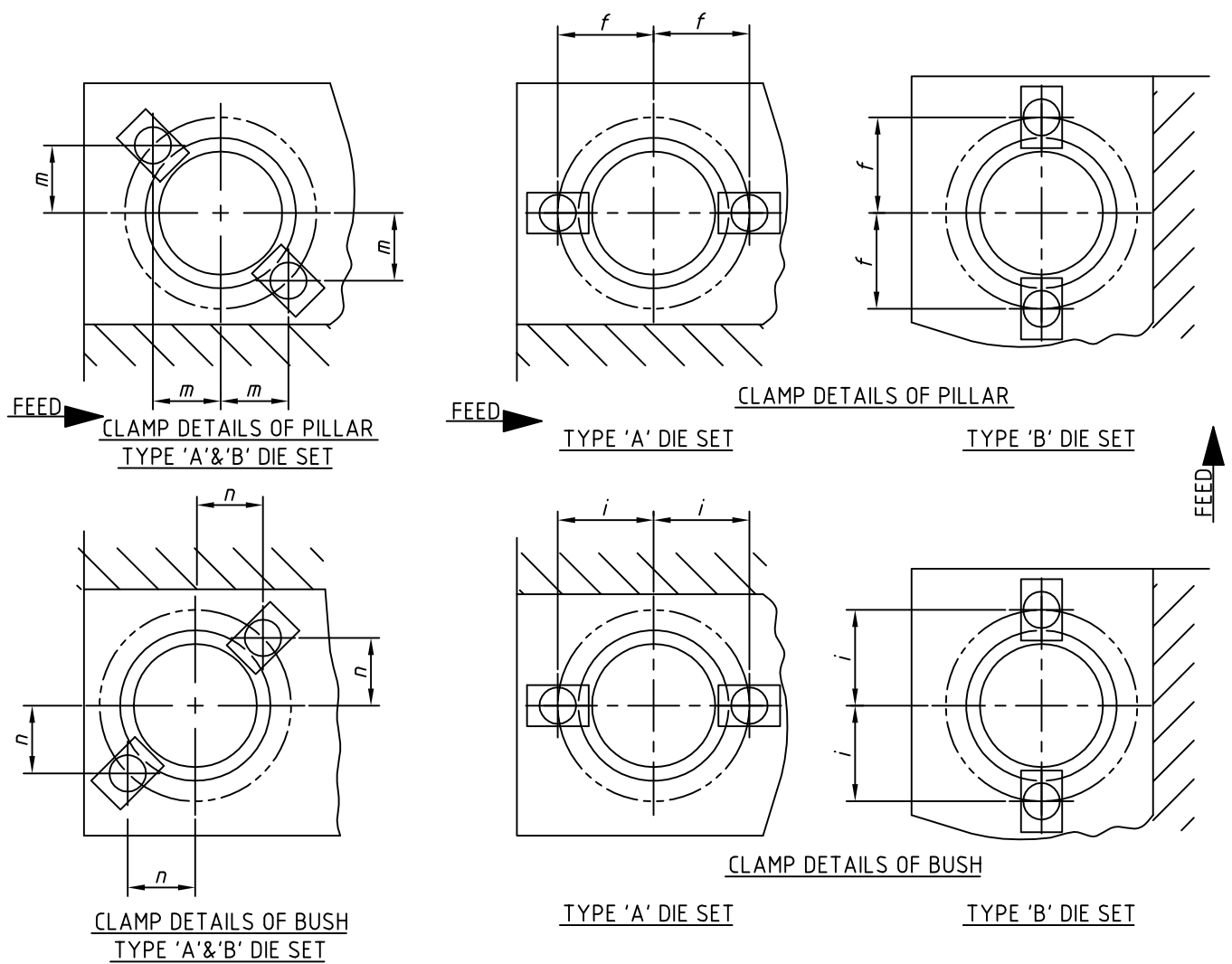
CLAMP ARRANGEMENT

05-03-23-4,0-1



SERIES	a	b	c	d	e	Ød1
2025	48	36	46	22	30	30/32
2030	48	36	46	22	30	30/32
2035	48	36	46	22	30	30/32
3030	48	36	46	22	30	30/32
3035	48	36	46	22	30	30/32
3035	62	46	57	32	40.5	50/52
3040	55	41	51	29.5	34.5	40/42
3040	62	46	57	32	40.5	50/52
3045	55	41	51	29.5	34.5	40/42
3045	62	46	57	32	40.5	50/52
3050	62	46	57	32	40.5	50/52
4040	55	41	51	29.5	34.5	40/42
4040	62	46	57	32	40.5	50/52
4045	62	46	57	32	40.5	50/52
4050	62	46	57	32	40.5	50/52
4060	62	46	57	32	40.5	50/52

DRN	01/10/09	SHAMNA MOHAN		
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			CLAMP ARRANGEMENT	05-03-23-4,0-2



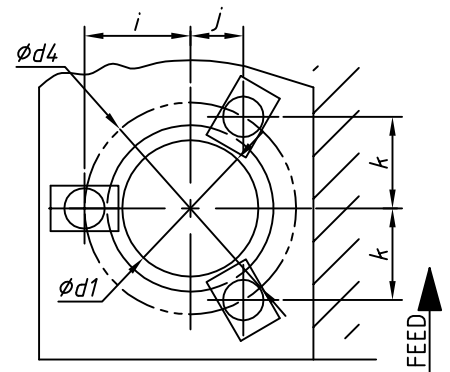
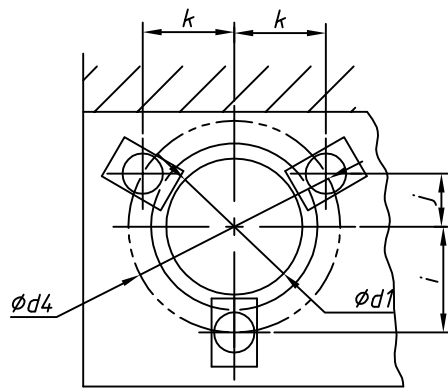
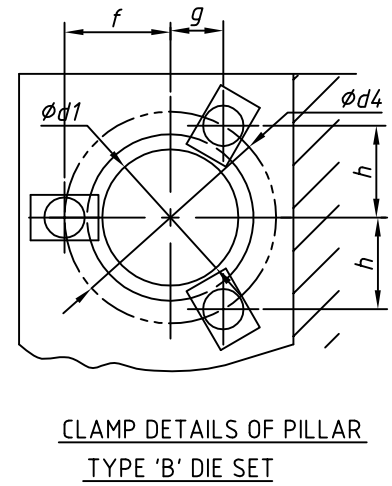
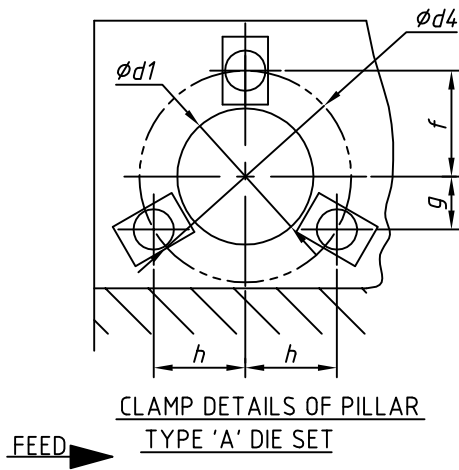
CO-ORDINATES OF CLAMPING
SCREWS FOR GUIDE PILLAR

$\phi d1$	$\phi d4$	f	m
18/19	34.0	17.0	12.0
24/25	46.0	23.0	16.3
30/32	52.0	26.0	18.4
40/42	62.0	31.0	21.9
50/52	72.0	36.0	25.5

CO-ORDINATES OF CLAMPING
SCREWS FOR GUIDE BUSHES

$\phi d1$	$\phi d4$	i	n
18/19	50.0	25.0	17.7
24/25	64.0	32.0	22.6
30/32	72.0	36.0	25.5
40/42	82.0	41.0	29.0
50/52	94.0	47.0	33.2

DRN	01/10/09	SHAMNA MOHAN	
CHD	-	-	
APPR.	-	-	
NTTF BANGALORE			CO-ORD CLAMP SCREWS (FOR GUIDE PILLAR&BUSH)
			05-03-23-4,0-3



CO-ORDINATES OF CLAMPING
SCREWS FOR GUIDE PILLAR

$\phi d1$	$\phi d4$	f	g	h
18/19	34.0	17.0	8.50	14.7
24/25	46.0	23.0	11.5	19.9
30/32	52.0	26.0	13.0	22.5
40/42	62.0	31.0	15.5	26.8
50/52	72.0	36.0	18.0	31.2

CO-ORDINATES OF CLAMPING
SCREWS FOR GUIDE BUSHES

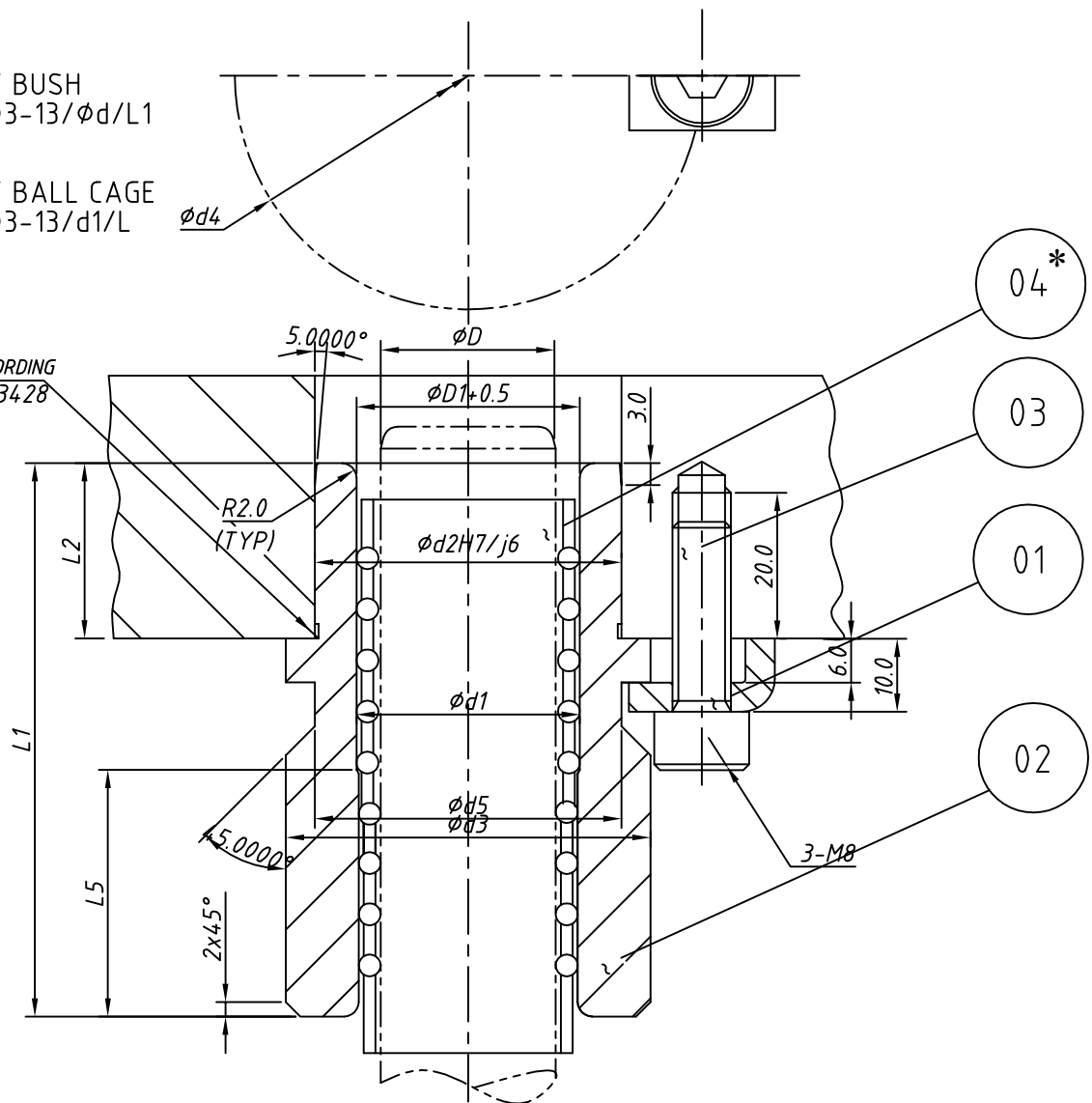
$\phi d1$	$\phi d4$	i	j	k
18/19	50.0	25.0	12.5	21.7
24/25	64.0	32.0	16.0	27.7
30/32	72.0	36.0	13.0	31.2
40/42	82.0	41.0	20.5	35.5
50/52	94.0	47.0	23.5	40.7

DRN	01/10/09	SHAMNA MOHAN	
CHD	-	-	
APPR.	-	-	
NTTF BANGALORE			CO-ORD CLAMP SCREWS (FOR GUIDE PILLAR&BUSH)
			05-03-23-4,0-4

TO SPECIFY BUSH
WRITE 05-03-13/ ϕ d/L1

TO SPECIFY BALL CAGE
WRITE 05-03-13/d1/L

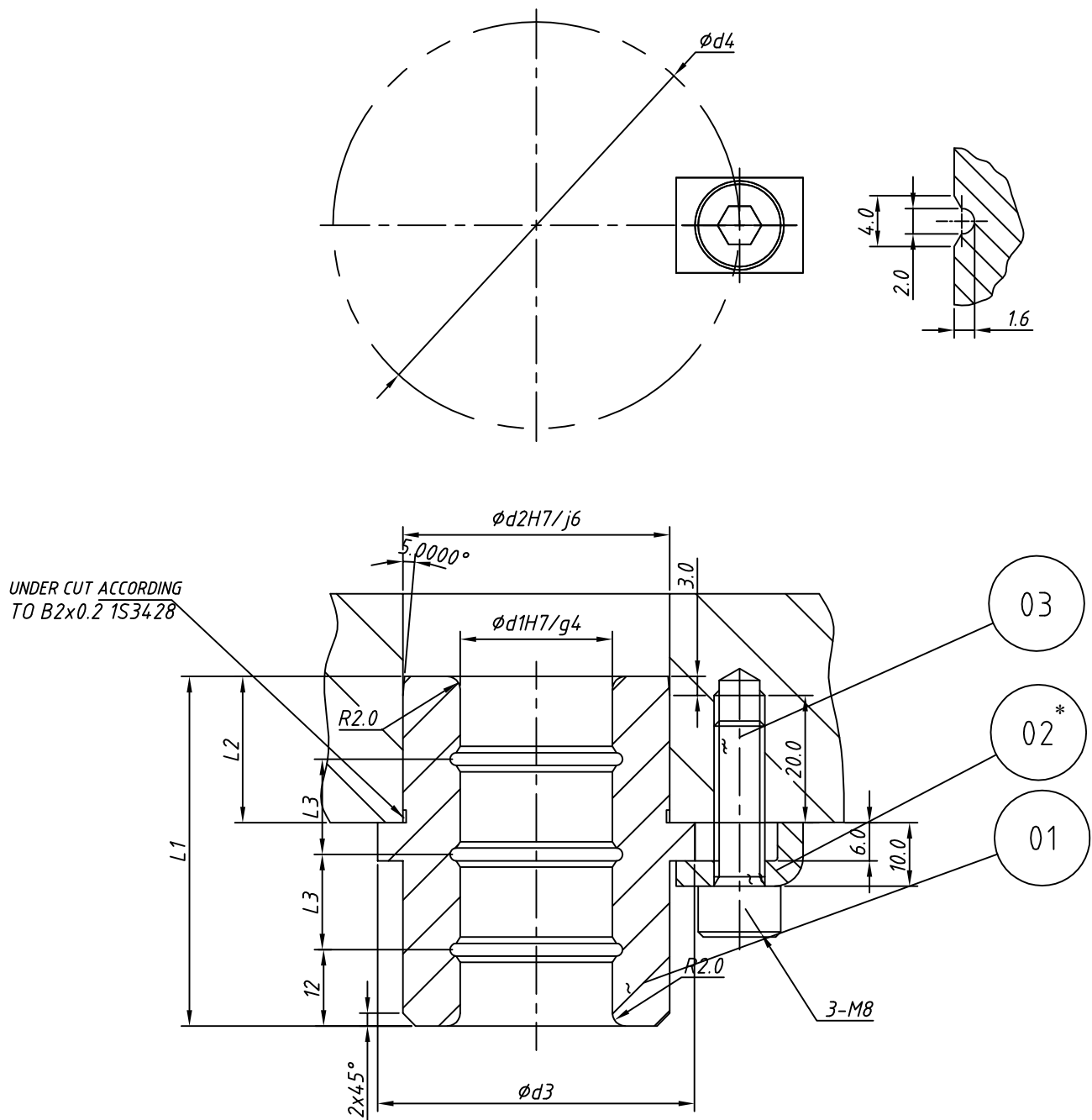
UNDER CUT ACCORDING
TO B2x0.2 1S3428



BALL CAGE				GUIDE BUSH							
PILLAR ϕ d	BALL ϕ	d1	LENGTH L	d1	d2	d3	d4	d5	L1	L2	L5
24/25	3	30/31	50-100	30/31	42	50	64	42	90	24	48
30/32	4	38/40	60-100	38/40	50	58	72	50	90	30	60
30/32	4	38/40	60-100	38/40	50	58	72	50	110	30	60
40/42	4	48/50	78-100	48/50	60	68	82	60	90	40	80
40/42	4	48/50	78-100	48/50	60	68	82	60	110	40	80
40/42	4	48/50	78-100	48/50	60	68	82	60	125	40	80
50/42	4	58/60	78-100	58/60	72	80	94	72	90	50	100
50/52	4	58/60	78-100	58/60	72	80	94	72	110	50	100

^x REFER 05-03 22

			4	BALL CAGE	1	STD			
			2	CLAMP	2	MS			
DRN	01/10/09	SHAMNA MOHAN	3	SHCS	2		M8	25	
CHD	-	-	1	GUIDE BUSH	1	E230			CHDN
APPR.	-	-	POS.	DESCRIPTION	QTY.	MAT.	FINISHED SIZE	LENGTH	REMARKS
NTTF BANGALORE			CO-ORD CLAMP SCREWS (FOR GUIDE PILLAR&BUSH)					05-03-13	



TO SPECIFY WRITE
05-03-17/ $\phi d1/L1$

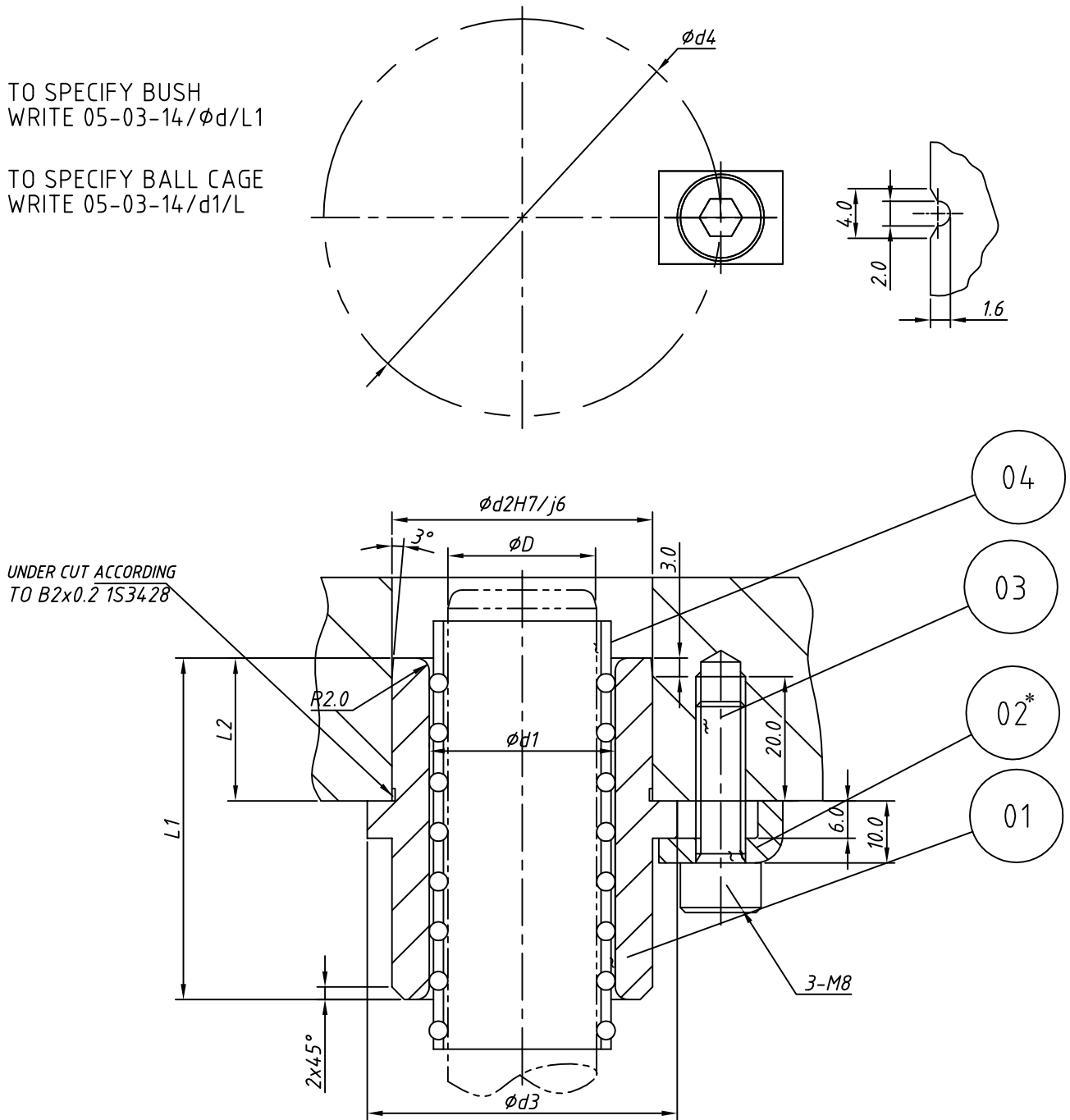
* REFER 05-03 22

$\phi d1$	$\phi d2$	$\phi d3$	$\phi d4$	L1	L2	L3	a	b	c
24/25	42	50	64	35/55	23	15	2	4.5	1.6
30/32	50	58	72	45/55	28	18	2	4.5	1.6
40/42	60	68	82	60	38	20	2	4.5	1.6
50/52	72	80	94	70	48	25	2	4.5	1.6

DRN	01/10/09	SHAMNA MOHAN	3	SHCS	3		M8	25	
CHD	-	-	2	CLAMP	3	MS			
APPR.	-	-	1	GUIDE BUSH	1	E230			CHDN
			POS.	DESCRIPTION	QTY.	MAT.	FINISHED SIZE	LENGTH	REMARKS
NTTF BANGALORE			DEMOUNTABLE GUIDE BUSH				05-03-17		

TO SPECIFY BUSH
WRITE 05-03-14/ ϕ d/L1

TO SPECIFY BALL CAGE
WRITE 05-03-14/d1/L

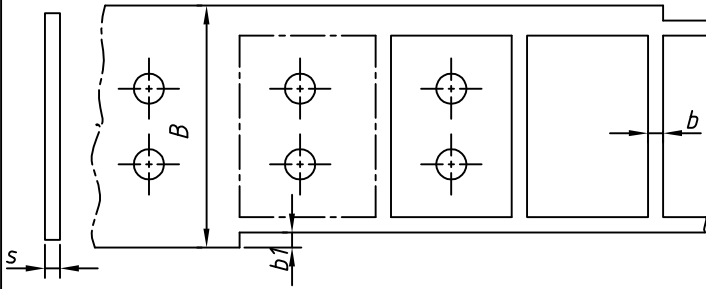


BALL CAGE				GUIDE BUSH					
PILLAR ϕd	BALL ϕ	d1	LENGTH L	d1	d2	d3	d4	L1	L2
24/25	3	30/31	50-100	30/31	42	50	64	35/55	23
30/32	4	38/40	60-100	38/40	50	58	72	45/55	28
40/42	4	48/50	78-100	48/50	60	68	82	60	38
50/52	4	58/60	78-100	58/60	72	80	94	75	46

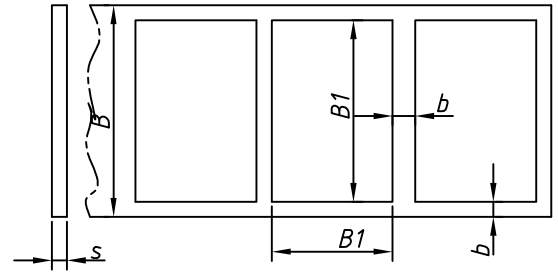
* REFER 05-03 22

DRN	01/10/09	SHAMNA MOHAN	4	BALL CAGE	1	STD			
CHD	-	-	3	SHCS	3		M8	25	
APPR.	-	-	2	CLAMP	3	MS			
			1	GUIDE BUSH	1	E230			CHDN
			POS.	DESCRIPTION	QTY.	MAT.	FINISHED SIZE	LENGTH	REMARKS
NTTF BANGALORE			DEMOUNTABLE GUIDE BUSH					05-03-14	

STRIP LAYOUT WITH PITCHCUTTER



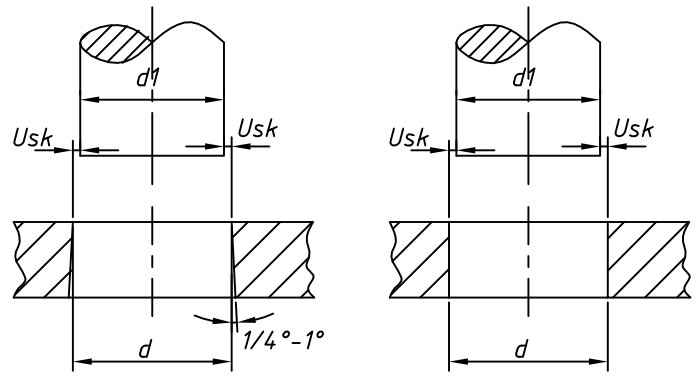
STRIP LAYOUT WITHOUT PITCH CUTTER



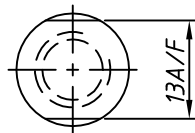
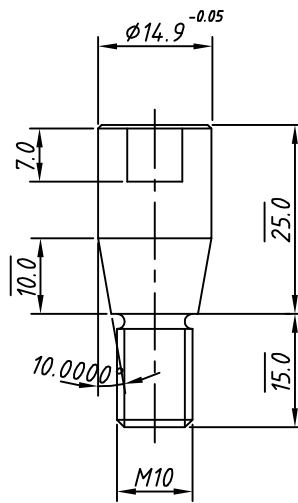
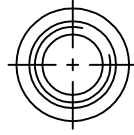
SHEET THICKNESS (S)	STRIP LAYOUT WITH PITCH CUTTER			
	B=20	B=50	B=75	B=100
0.10	1.0	1.2	1.5	1.9
0.18	1.0	1.2	1.5	1.9
0.20	1.0	1.2	1.5	1.9
0.22	1.0	1.3	1.6	2.0
0.24	1.0	1.3	1.6	2.0
0.28	1.0	1.3	1.6	2.0
0.32	1.0	1.4	1.7	2.1
0.38	1.0	1.4	1.7	2.1
0.44	1.0	1.4	1.7	2.1
0.50	1.2	1.5	1.8	2.2
0.56	1.2	1.5	1.8	2.2
0.63	1.3	1.6	1.9	2.3
0.75	1.3	1.6	1.9	2.3
0.88	1.4	1.7	2.0	2.4
1.00	1.5	1.8	2.0	2.4
1.13	1.5	1.9	2.1	2.4
1.25	1.6	2.0	2.2	2.6
1.38	1.7	2.1	2.3	2.8
1.50	1.8	2.2	2.5	3.0
1.75	2.1	2.5	2.8	3.2
2.00	2.4	2.8	3.0	3.3
2.25	2.6	3.0	3.3	3.6
2.50	2.8	3.3	3.6	4.0
2.75	3.0	3.5	4.0	4.5
3.0				
3.5				
4.0				
4.5				
4.75				
5				
6				
7				
8				
9				
10				

STRIP LAYOUT WITHOUT PITCH CUTTER							
B=10	B=50	B=100	B=150	B=250	B=350	B=500	B=1000
1.2							
1.2	1.5						
1.2	1.5	1.8					
1.2	1.6	1.9					
1.3	1.6	2.0	2.5				
1.3	1.7	2.0	2.7				
1.3	1.7	2.4	2.9	3.3			
1.4	1.8	2.6	3.1	3.5			
1.4	1.9	2.8	3.3	3.7	4.0		
1.5	2.0	3.0	3.5	4.0	4.5		
1.4	1.9	2.8	3.3	3.8	4.3	5.0	
1.3	1.8	2.6	3.1	3.6	4.1	4.6	
1.2	1.7	2.4	2.9	3.4	3.9	4.4	
1.1	1.6	2.2	2.7	3.2	3.7	4.2	
1.0	1.5	2.0	2.5	3.0	3.5	4.0	6.0
1.2	1.7	2.2	2.7	3.2	3.7	4.2	6.0
1.4	1.9	2.4	2.9	3.4	3.9	4.3	6.0
1.5	1.9	2.5	3.0	3.5	4.0	4.5	6.2
1.6	2.0	2.5	3.0	3.5	4.0	4.5	6.5
1.8	2.2	2.7	3.2	3.7	4.2	4.7	6.7
2.0	2.5	3.0	3.5	4.0	4.5	5.0	7.0
2.0	2.8	3.2	3.7	4.2	4.7	5.2	7.2
2.0	3.0	3.5	4.0	4.5	5.0	5.5	7.5
2.0	3.2	3.7	4.2	4.7	5.2	5.7	7.7
2.0	3.5	4.0	4.5	5.0	5.5	6.0	8.0
2.5	3.7	4.2	4.7	5.2	5.7	6.5	8.0
2.5	4.0	4.5	5.0	5.5	6.0	7.0	8.0
3.0	4.2	4.7	5.2	5.7	6.2	7.0	8.5
3.0	4.5	4.7	5.2	5.7	6.2	7.0	8.5
3.0	4.5	5.0	5.5	6.0	6.5	7.0	9.0
3.5	5.0	5.5	6.0	6.5	7.0	8.0	9.5
4.0	5.5	6.0	6.5	7.0	8.0	9.0	10.0
5.0	6.0	6.5	7.0	8.0	9.0	10.0	11.0
6.0	7.0	7.5	8.0	9.0	10.0	11.0	12.0
7.0	8.0	8.5	9.0	10.0	11.0	12.0	13.0

DRN	01/10/09	SHAMNA MOHAN		
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			- -	-



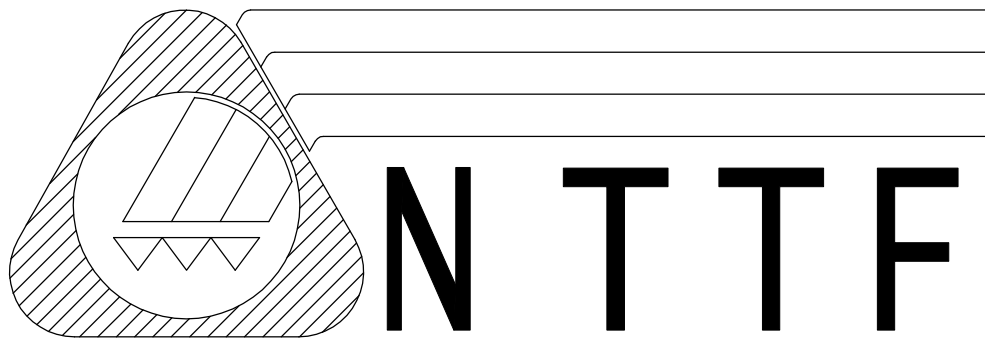
DRN	01/10/09	SHAMNA MOHAN		
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			- -	-



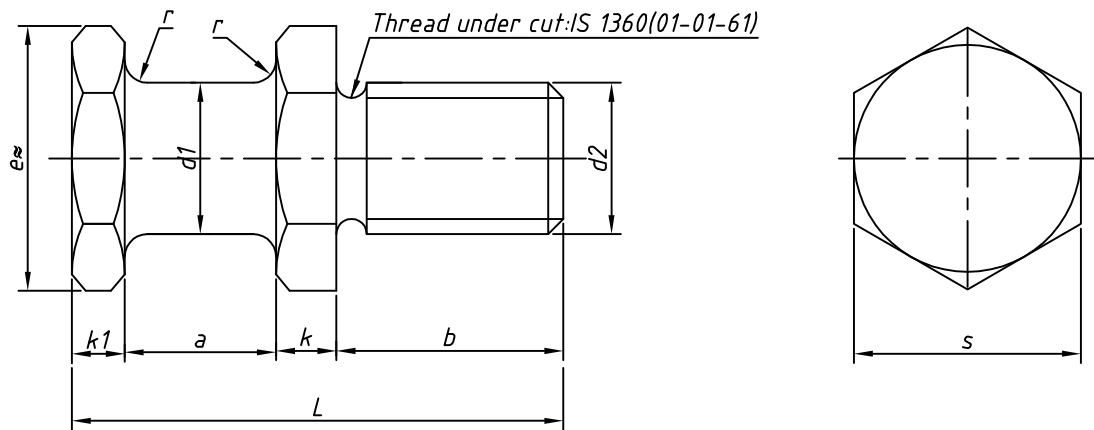
TO SPECIFY WRITE
05-03-24

DRN	01/10/09	SHAMNA MOHAN	MATERIAL:MS	
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			SHANK FOR GENENCO TOGGLE PRESS	05-03-24

GENERAL STANDARDS

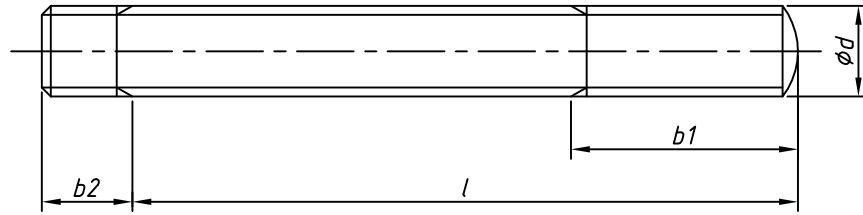


**COMPILED BY SRIVINAY
PGDTD 2009,
NTTF BANGALORE**



d_1	d_2	a	b	k	k_1	L	s	e	r	LIFTING FORCE
16	M16x1.5	16	24	5	5	50	24	28	2.5	4KN(400kg)
20	M20x1.5	20	30	8	7	65	30	35	3	6.5KN(650kg)
25	M24x1.5	25	36	10	9	80	36	42	3	10KN(1000kg)
32	M30x1.5	32	45	13	10	100	46	54	4	15KN(1500kg)
40	M36x2	40	54	14	12	120	55	64	4	22KN(2200kg)
48	M42x2	48	63	15	14	140	65	75	4	30KN(3000kg)

DRN	01/10/09	SHAMNA MOHAN			
CHD	-	-			
APPR.	-	-			
NTTF BANGALORE			CARRIER SCREW		04-00-01

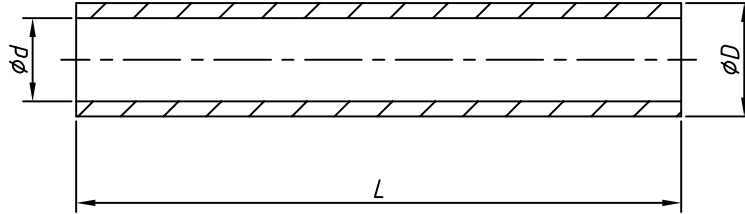


d Nominal size h13	$b \pm \frac{p}{2}$			$b_2 \text{ js } 16$			Preferred Length For Type A & Type B		Preferred Length For Type C	
	$l \leq 125$	$125 < l < 200$	$l > 200$	TYP A	TYP B	TYP C	MIN	MAX	MIN	MAX
M3	12			3	4.5		14	25		
M4	14			4	6	8	16	70	25	60
M5	16			5	7.5		20	80		
M6	18			6	9		25	85		
M8	22			8	12	16	30	110	40	100
M10	26	32		10	15		30	150		
M12	30	36		12	18	24	35	170	80	140
M16	38	44	57	16	24	32	45	300	100	160
M20	46	52	65	20	30	40	55	300	120	200
M24	54	60	73	24	36		65	300		
M30	66	72	85	30	45		75	300		
M36	78	84	97	36	54		90	300		

PREFERRED LENGTH: 14, 16, 20, 25, 30, 35, 40, 45, 50, 55, 60, 65, 70, 75, 80, 85, 90, 100, 110, 120, 130, 140, 150, 160, 170, 180, 190, 200, 225, 250, 275, 300.

TO SPECIFY : TYPE A, $d=M5$, $l=25$, WRITE N0:04-00-02/ d/l

DRN	01/10/09	SHAMNA MOHAN			
CHD	-	-			
APPR.	-	-			
NTTF BANGALORE			STUD		04-00-02

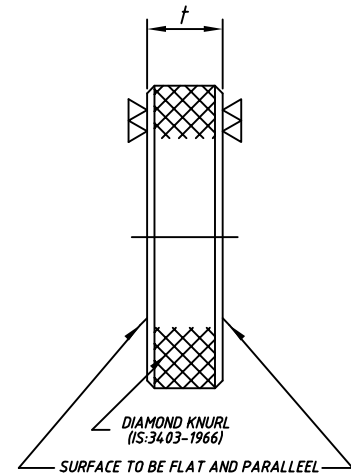
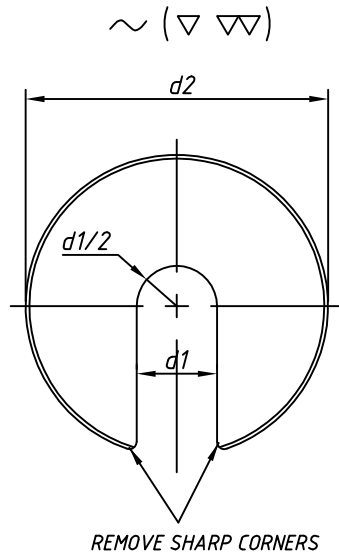


S NO	To suit	d	D
1	M5	6	10
2	M6	7	10
3	M8	9	12
4	M10	11	15
5	M12	14	16
6	M14	16	20
7	M16	18	22

TO SPECIFY : WRITE 04-00-30/d/l

AVAILABLE AT M/S INTERNATIONAL TOOLS

DRN	01/10/09	SHAMNA MOHAN		
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			SPACER BUSH	04-00-30



$d1$ H12	t	$d2$							For Bolt or Screw Size
6.4	6	25	32						M6
8.4	8		32	40	22				M8
10.5	10			40	50				M10
13	10			22	50	63	22		M12
17	12			22	22	63	80		M16
21	12					63	80		M20
25	12						80	100	M24
31	15						80	100	M30

NOTE:

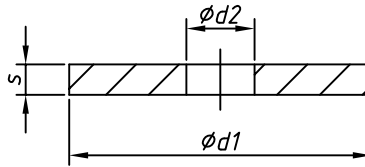
THE DIMENSION $d2$ IS BEFORE KNURLING

HARDNESS :30-40 HRC (CASE HARDENED)

TO SPECIFY : C WASHER OF $d1=13\text{mm}$ AND $d2=50\text{mm}$

WRITE NO :04-00-31/ $d1/d2$

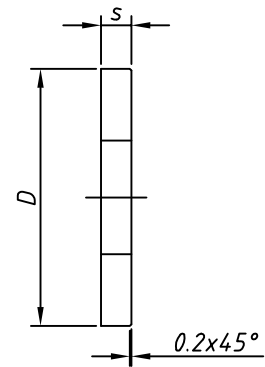
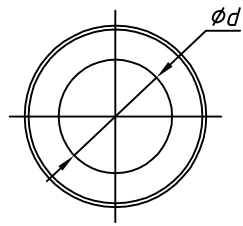
DRN	01/10/09	SHAMNA MOHAN	ACCORDING IS-4291	
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			C-WASHER	04-00-31



$d1$	$d2$	s
26	6.6	3
32	6.6	3
40	9.0	4
54	11.0	4
54	13.0	4

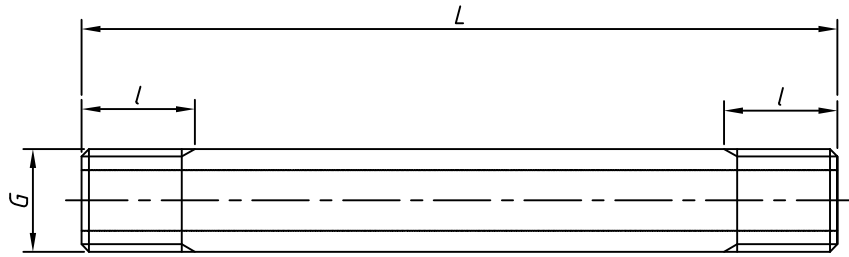
TO SPECIFY : C WASHER OF $d2=9.0\text{mm}$
 SHOULD DESIGNATED AS 04-00-32- $d2=9.0/d1=40$
 GEN.TOL.= ± 0.2

DRN	01/10/09	SHAMNA MOHAN	MATERIAL:MS	
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			WASHER	04-00-32



SIZE H12 d	D		S		FOR BLOT OR SCREW SIZE
	BASIC	TOLE.	BASIC	TOLE.	
2.2	5	$\begin{smallmatrix} +0 \\ -0.3 \end{smallmatrix}$	0.3	± 0.1	M2
3.2	7	$\begin{smallmatrix} +0 \\ -0.3 \end{smallmatrix}$	0.5	± 0.1	M3
4.3	9	$\begin{smallmatrix} +0 \\ -0.3 \end{smallmatrix}$	0.8	± 0.1	M4
5.3	10	$\begin{smallmatrix} +0 \\ -0.3 \end{smallmatrix}$	1.0	± 0.1	M5
6.4	12.5	$\begin{smallmatrix} +0 \\ -0.4 \end{smallmatrix}$	1.6	± 0.2	M6
8.4	17	$\begin{smallmatrix} +0 \\ -0.4 \end{smallmatrix}$	1.6	± 0.2	M8
10.5	21	$\begin{smallmatrix} +0 \\ -0.5 \end{smallmatrix}$	2	± 0.2	M10
13	24	$\begin{smallmatrix} +0 \\ -0.5 \end{smallmatrix}$	2.5	± 0.3	M12
15	28	$\begin{smallmatrix} +0 \\ -0.5 \end{smallmatrix}$	2.5	± 0.3	M14
17	30	$\begin{smallmatrix} +0 \\ -0.5 \end{smallmatrix}$	3	± 0.3	M16
21	37	$\begin{smallmatrix} +0 \\ -0.8 \end{smallmatrix}$	3	± 0.3	M20
23	39	$\begin{smallmatrix} +0 \\ -0.8 \end{smallmatrix}$	3	± 0.3	M22
25	44	$\begin{smallmatrix} +0 \\ -0.8 \end{smallmatrix}$	4	± 0.3	M24

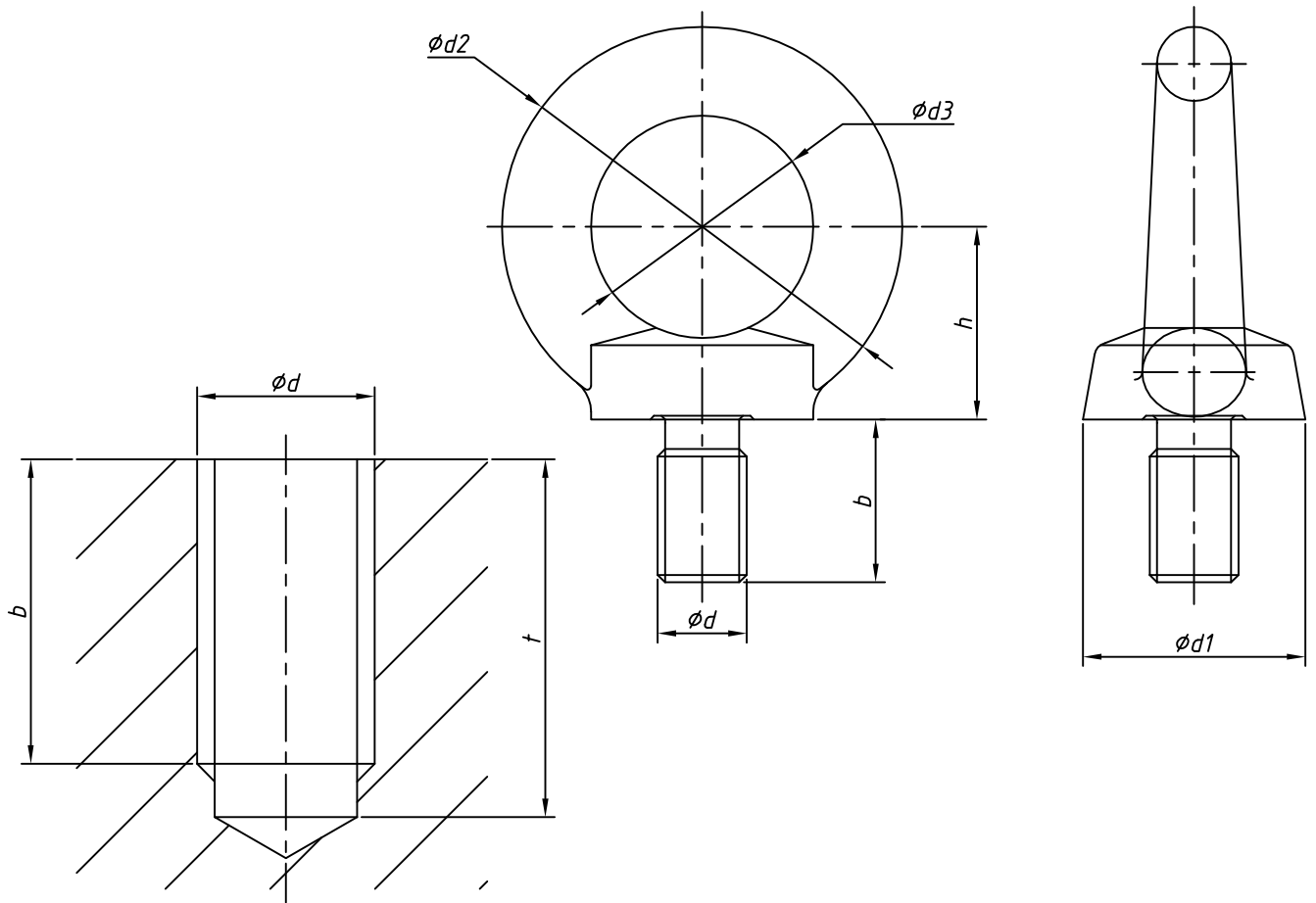
DRN	01/10/09	SHAMNA MOHAN		
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			WASHER	04-00-33



<i>G</i>	<i>l</i>	<i>L</i>
<i>G1/8"</i>	<i>10</i>	<i>50</i>
		<i>100</i>
		<i>150</i>
		<i>200</i>
<i>G1/4"</i>	<i>10</i>	<i>50</i>
		<i>100</i>
		<i>150</i>
		<i>200</i>
<i>G3/8"</i>	<i>15</i>	<i>100</i>
		<i>150</i>
		<i>200</i>
<i>G1/2"</i>	<i>15</i>	<i>100</i>
		<i>150</i>
		<i>200</i>

FOR ORDERING 1/8"x50
WRITE 04-02-01/12-3G1/8"x50
MATL:G.I.PIPE

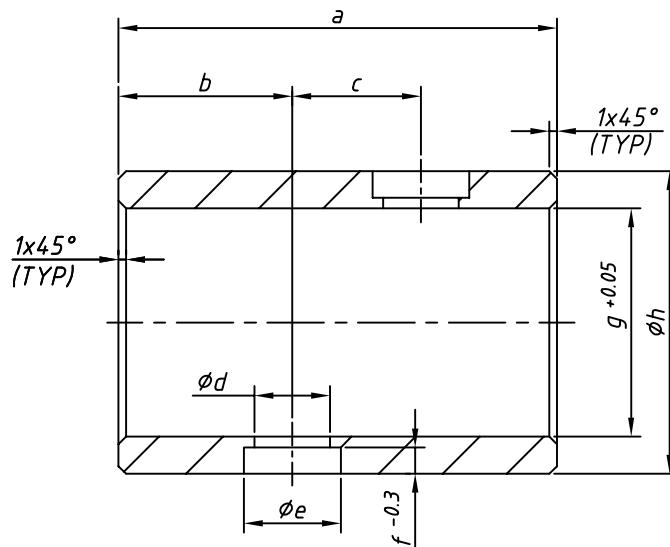
DRN	01/10/09	SHAMNA MOHAN		
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			EXTENSION PIPE	04-02-01/1,2-3



<i>d</i>	<i>M10</i>	<i>M12</i>	<i>M16</i>	<i>M20</i>	<i>M24</i>	<i>M30</i>	<i>M36</i>	<i>M42</i>
<i>b</i>	18	22	28	30	38	45	55	65
<i>d1</i>	25	30	35	40	50	65	75	85
<i>d2</i>	45	54	63	72	90	108	126	144
<i>d3</i>	25	30	35	40	50	60	70	80
<i>h</i>	22	26	30	35	45	55	65	75
<i>t</i>	24	29	35	38	47	55	66	77

REFER: IS 4190-1961

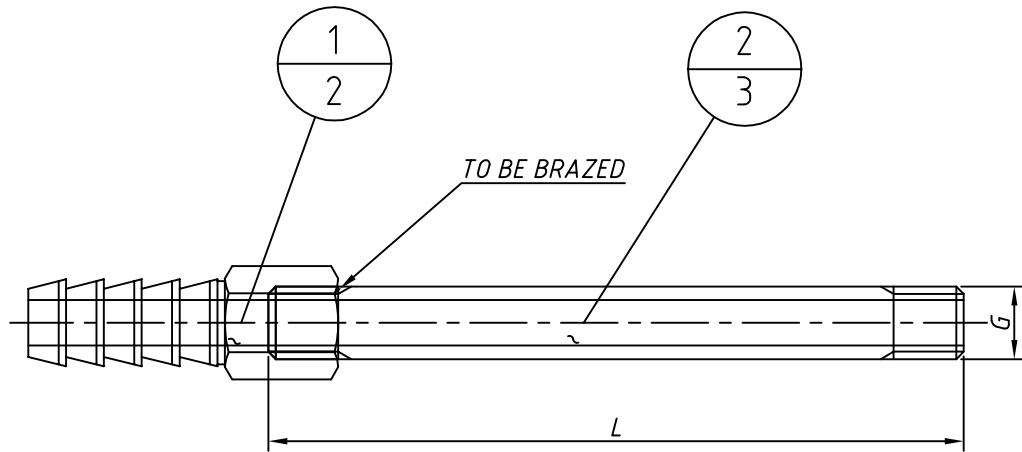
DRN	01/10/09	SHAMNA MOHAN			
CHD	-	-			
APPR.	-	-			
NTTF BANGALORE			EYE BOLT		04-02-03



BSP	a	b	c	d	e	f	g	h
1/4"	22	28	30	38	45	55	65	65
3/8"	30	35	40	50	65	75	85	85
1/2"	54	63	72	90	108	126	144	144

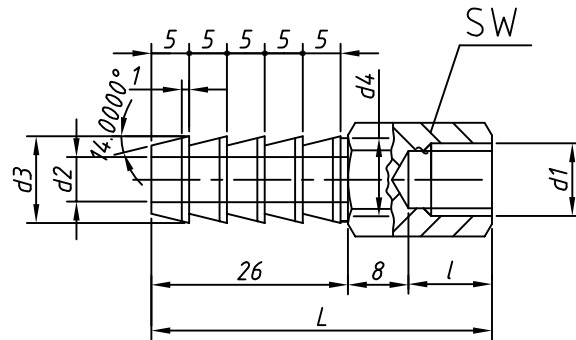
REFER: IS 4190-1961

DRN	01/10/09	SHAMNA MOHAN		
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			EYE BOLT	04-02-03



FOR ORDERING WRITE 04-02 01/1,2-1-GxL

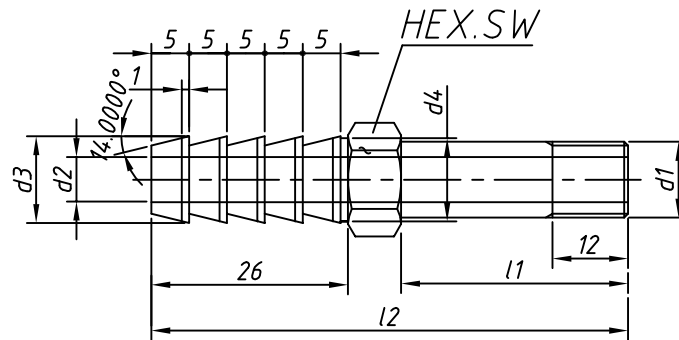
DRN	01/10/09	SHAMNA MOHAN	2	EXTENSION PIPE1					
CHD	-	-	1	CONNECTOR PLUG					CHDN
APPR.	-	-	POS.	DESCRIPTION	QTY.	MAT.	FINISHED SIZE	LENGTH	REMARKS
NTTF BANGALORE				WATER CONNECTOR			04-02-01/1,2-1		



$d1$	$d2$	$d3$	$d4$	SW	l	L
G1/8"	6.0	11.5	11.0	13.0	13.0	47.0
G1/4"	8.0	13.5	12.5	17.0	13.0	47.0
G3/8"	10.0	15.3	14.5	19.0	18.0	52.0

FOR ORDERING WRITE 04-02 01/1,2-2-d1

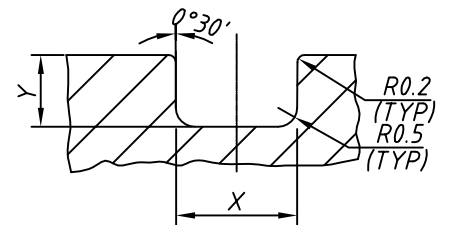
DRN	01/10/09	SHAMNA MOHAN	MATERIAL:BRASS	
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			CONNECTOR PLUG	04-02-01/1,2-2



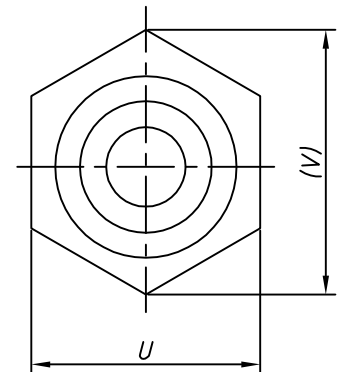
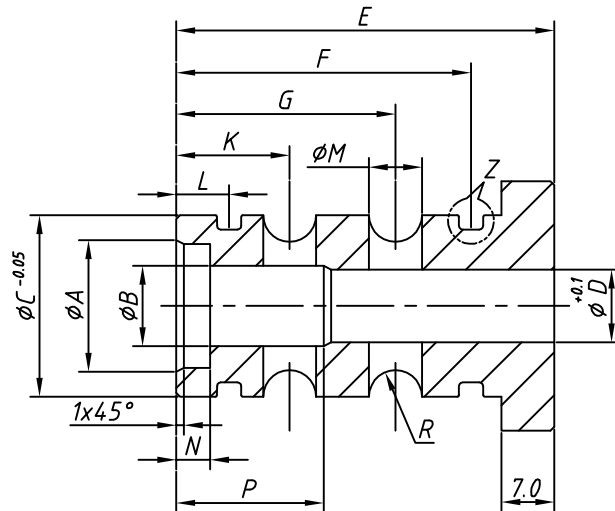
$d1$	$d2$	$d3$	$d4$	$l1$	$l2$	SW
$M10 \times 1$	6	11.5	11	18	50	13
				32	64	
$1/8''BSP$	6	11.5	11	18	50	13
				32	64	
$1/4''BSP$	8	13.5	12.5	18	50	13
				32	64	
$3/8''BSP$	10	15.5	14.5	18	50	19
				32	64	

FOR ORDERING $1/4''BSP$ $l1=32$: WRITE 04-02-02/ $1/4'' \times 32$

DRN	01/10/09	SHAMNA MOHAN	MATERIAL:BRASS	
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			HOSE CONNECTOR	04-02-02

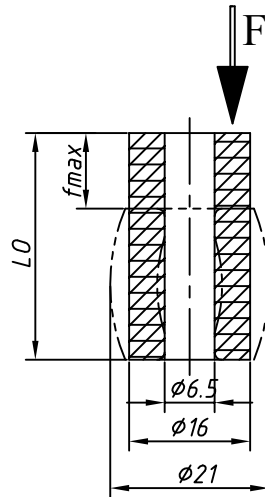


'O' RING GROOVE DETAIL
ENLARGED VIEW-Z



$\frac{W}{BSP}$	A	B	C	D	E	F	G	K	L	M	N	P	U	(V)	X	Y
1/4"	13.6	8.5	20	6.3	41.0	29.0	23.0	13.0	7.0	4.0	4.5	16.0	24	26.2	3.1	1.9
3/8"	17.4	10.5	24	9.5	50.0	37.0	29.0	15.0	7.0	7.0	4.5	19.5	30	33.0	3.1	1.9
1/2"	21.6	14.5	30	12.7	65.0	51.0	40.0	23.0	12.0	10.0	6.0	29.0	36	40.0	4.6	2.8

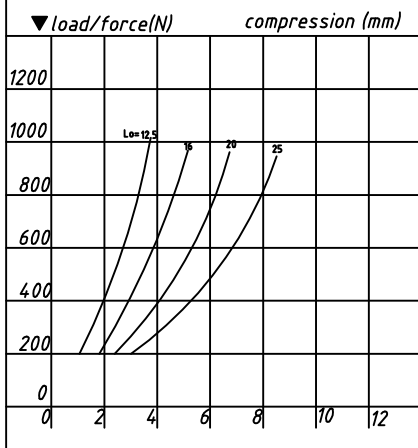
DRN	01/10/09	SHAMNA MOHAN				
CHD	-	-				
APPR.	-	-				
NTTF BANGALORE			CASCADE WATER HOSE CONNECTOR			04-02-05-1,3-2



SPRING dia=16mm-80 SHORE

Lo mm	fmax mm	Fmax N
12.5	4.3	1020
16	5.6	980
20	7	950
25	8.7	940

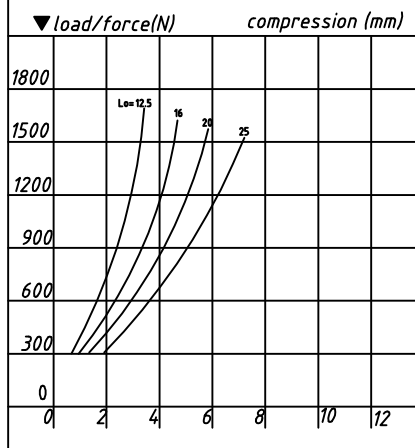
LOAD/DEFLECTION GRAPH



SPRING dia=16mm-90 SHORE

Lo mm	fmax mm	Fmax N
12.5	3.6	1680
16	4.8	1650
20	6	1620
25	7.5	1580

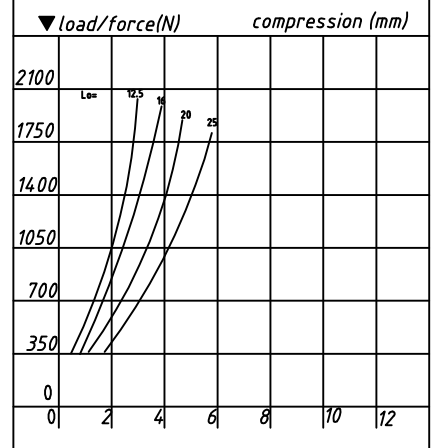
LOAD/DEFLECTION GRAPH



SPRING dia=16mm-95 SHORE

Lo mm	fmax mm	Fmax N
12.5	3.1	2000
16	4	1920
20	5	1900
25	6.2	1870

LOAD/DEFLECTION GRAPH



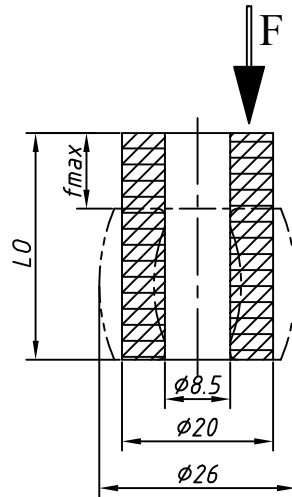
DRN	01/10/09	SHAMNA MOHAN
CHD	-	-
APPR.	-	-

REF:FIBROFLEX-ELASTOMERS

NTTF
BANGALORE

PUR SPRING

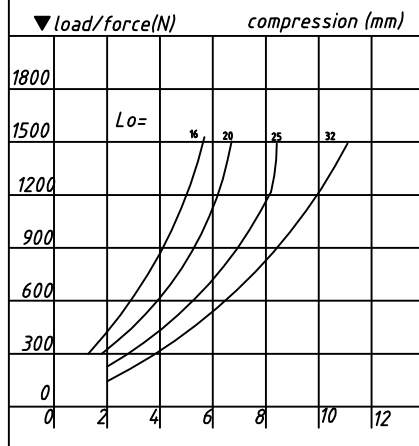
04-01-02-0,7-1



SPRING dia=20mm-80 SHORE

Lo mm	fmax mm	Fmax N
16	5.6	1530
20	7	1510
25	8.7	1500
32	10.6	1490

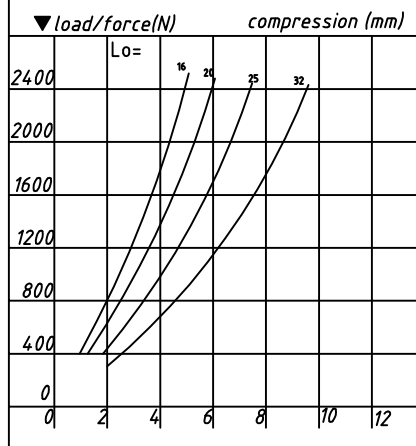
LOAD/DEFLECTION GRAPH



SPRING dia=20mm-90 SHORE

Lo mm	fmax mm	Fmax N
16	4.8	2600
20	6	2550
25	7.5	2530
32	9.6	2500

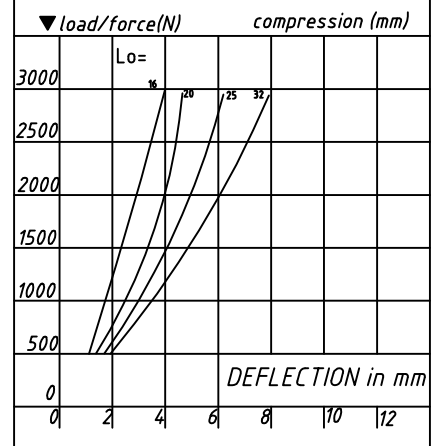
LOAD/DEFLECTION GRAPH



SPRING dia=20mm-95SHORE

Lo mm	fmax mm	Fmax N
16	4	3050
20	5	3000
25	6.2	2980
32	8	2950

LOAD/DEFLECTION GRAPH



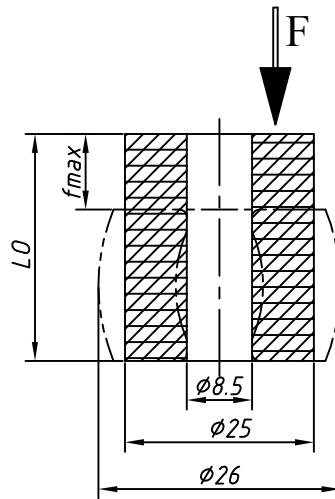
DRN	01/10/09	SHAMNA MOHAN
CHD	-	-
APPR.	-	-

REF:FIBROFLEX-ELASTOMERS

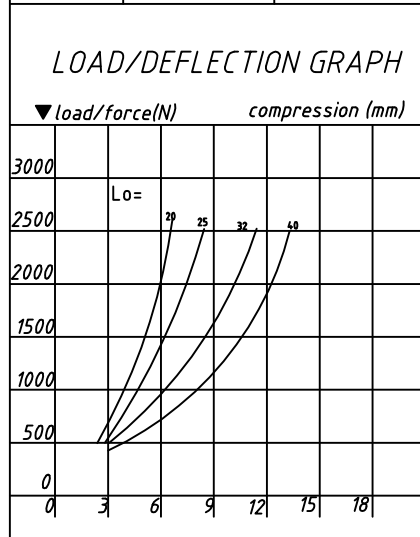
NTTF
BANGALORE

PUR SPRING

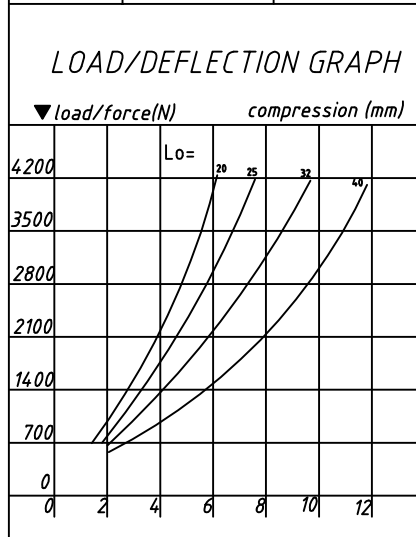
04-01-02-1,7-2



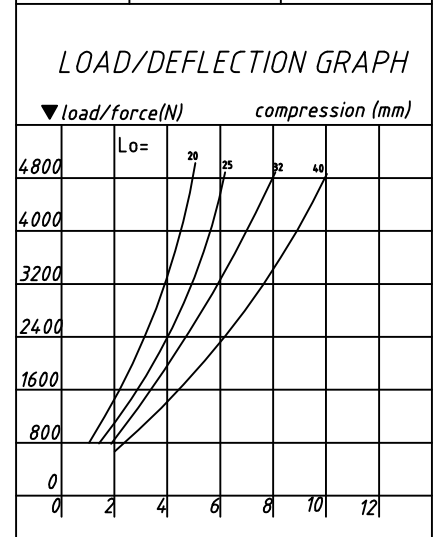
SPRING dia=25mm-80 SHORE		
Lo mm	fmax mm	Fmax N
20	7	2600
25	8.7	2550
32	10.6	2520
40	14	2500



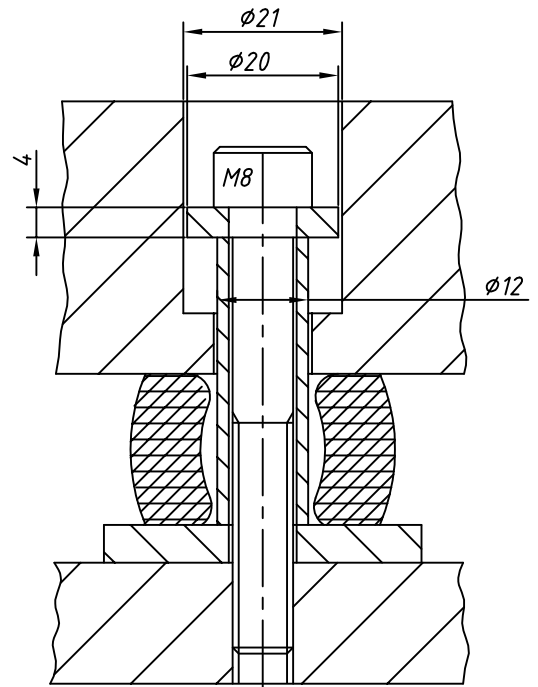
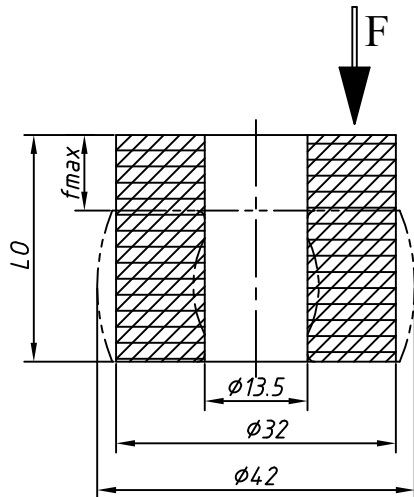
SPRING dia=25mm-90 SHORE		
Lo mm	fmax mm	Fmax N
20	6	4300
25	7.5	4200
32	9.6	4150
40	12	4120



SPRING dia=25mm-95 SHORE		
Lo mm	fmax mm	Fmax N
20	5	5100
25	6.2	5080
32	8	5020
40	10	5000



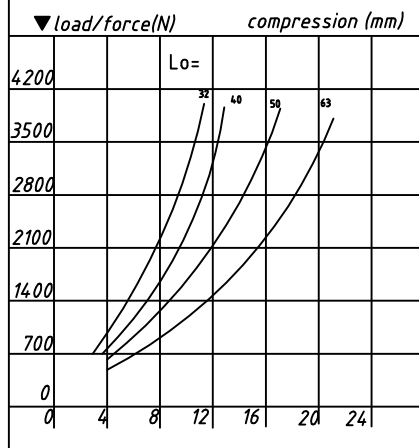
DRN	01/10/09	SHAMNA MOHAN	REF:FIBROFLEX-ELASTOMERS	
CHD	-	-		
APPR.	-	-		
NTTF BANGALORE			PUR SPRING	04-01-02-1,7-3



SPRING dia=32mm-80 SHORE

Lo mm	fmax mm	Fmax N
32	10.6	3900
40	14	3850
50	17.5	3820
63	22	3800

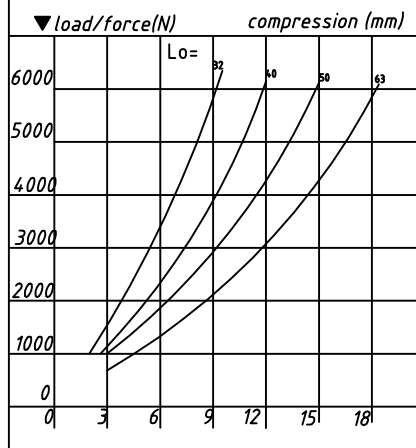
LOAD/DEFLECTION GRAPH



SPRING dia=32mm-90 SHORE

Lo mm	fmax mm	Fmax N
32	9.6	6400
40	12	6350
50	15	6300
63	18.9	6250

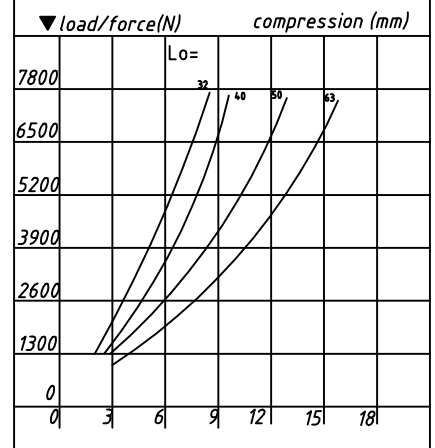
LOAD/DEFLECTION GRAPH



SPRING dia=32mm-95 SHORE

Lo mm	fmax mm	Fmax N
32	8	7600
40	10	7500
50	12	7480
63	15.7	7450

LOAD/DEFLECTION GRAPH



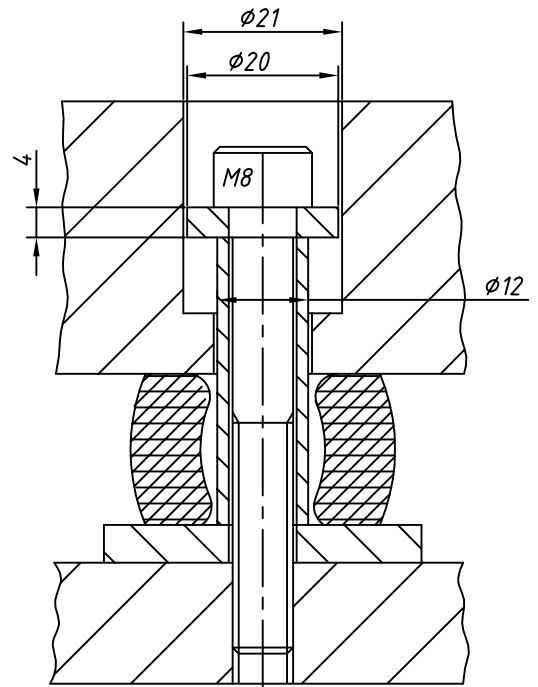
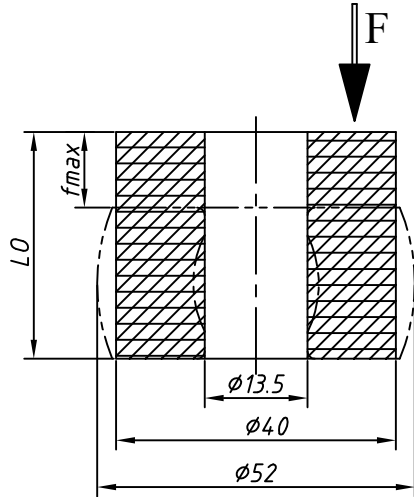
DRN	01/10/09	SHAMNA MOHAN
CHD	-	-
APPR.	-	-

REF:FIBROFLEX-ELASTOMERS

NTTF
BANGALORE

PUR SPRING

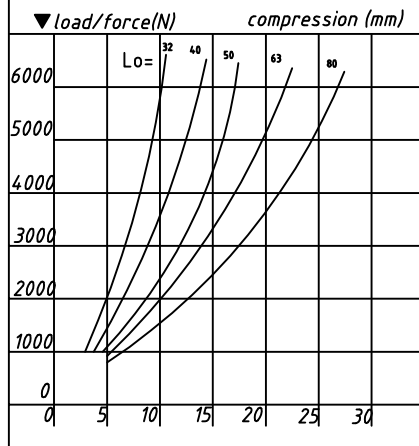
04-01-02-1,7-4



SPRING dia=40mm-80 SHORE

Lo mm	fmax mm	Fmax N
32	10.6	6700
40	14	6600
50	17.5	6550
63	22	6500
80	28	6480

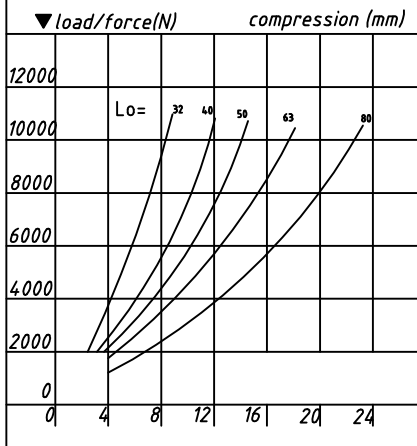
LOAD/DEFLECTION GRAPH



SPRING dia=40mm-90 SHORE

Lo mm	fmax mm	Fmax N
32	9.6	11000
40	12	10900
50	15	10800
63	18.9	10750
80	24	10700

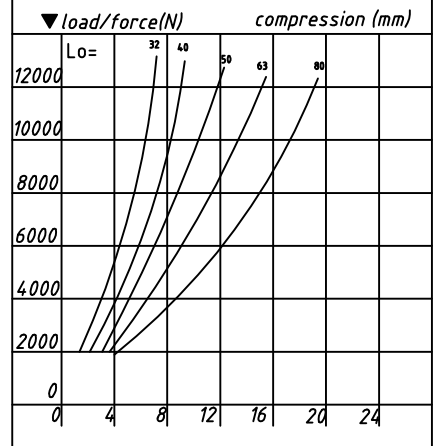
LOAD/DEFLECTION GRAPH



SPRING dia=40mm-95 SHORE

Lo mm	fmax mm	Fmax N
32	8	13000
40	10	12700
50	12	12500
63	15.7	12450
80	20	12430

LOAD/DEFLECTION GRAPH



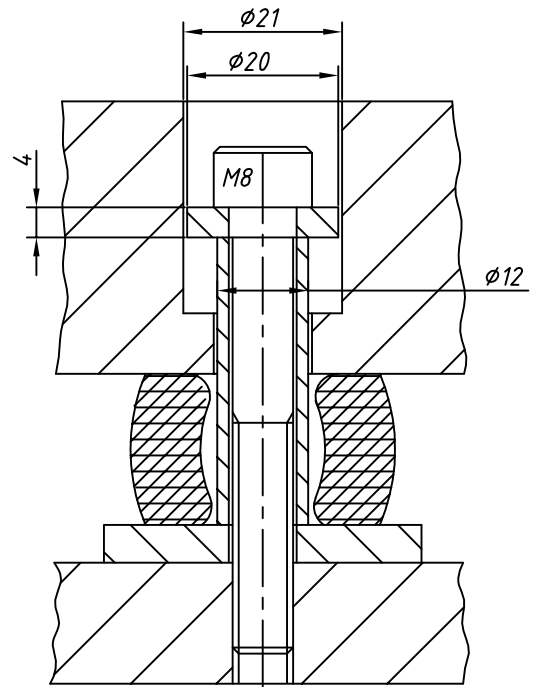
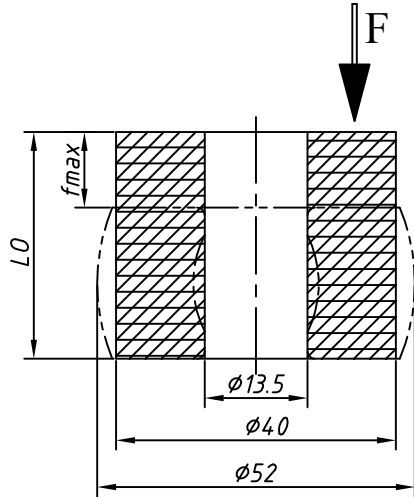
DRN	01/10/09	SHAMNA MOHAN
CHD	-	-
APPR.	-	-

REF:FIBROFLEX-ELASTOMERS

NTTF
BANGALORE

PUR SPRING

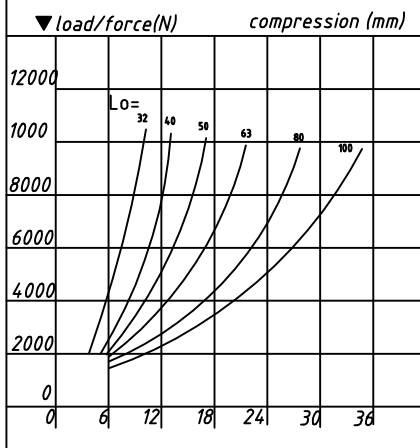
04-01-02-1,7-5



SPRING dia=50mm-80 SHORE

Lo mm	fmax mm	Fmax N
32	10.6	10800
40	14	10400
50	17.5	10200
63	22	10000
80	28	9950
100	35	9900

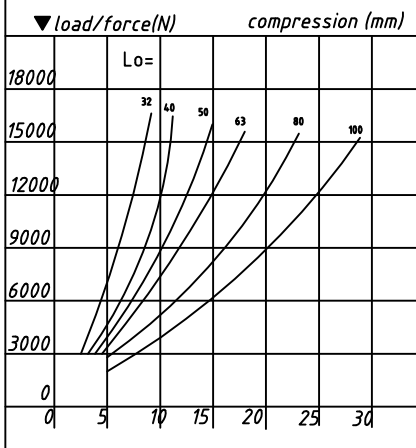
LOAD/DEFLECTION GRAPH



SPRING dia=50mm-90 SHORE

Lo mm	fmax mm	Fmax N
32	9.6	17400
40	12	17300
50	15	17000
63	18.9	16650
80	24	16500
100	30	16400

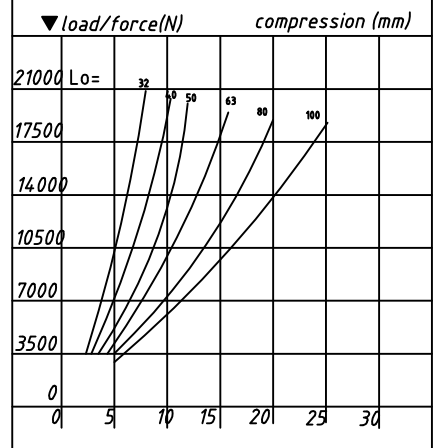
LOAD/DEFLECTION GRAPH



SPRING dia=50mm-95 SHORE

Lo mm	fmax mm	Fmax N
32	8	21000
40	10	20100
50	12	19600
63	15.7	19200
80	20	19100
100	25	19050

LOAD/DEFLECTION GRAPH



DRN	01/10/09	SHAMNA MOHAN
CHD	-	-
APPR.	-	-

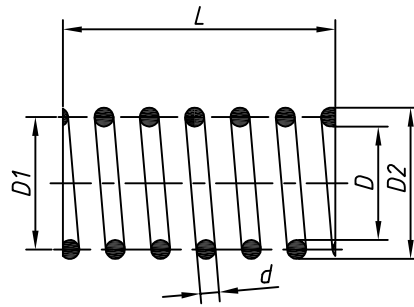
REF:FIBROFLEX-ELASTOMERS

NTTF
BANGALORE

PUR SPRING

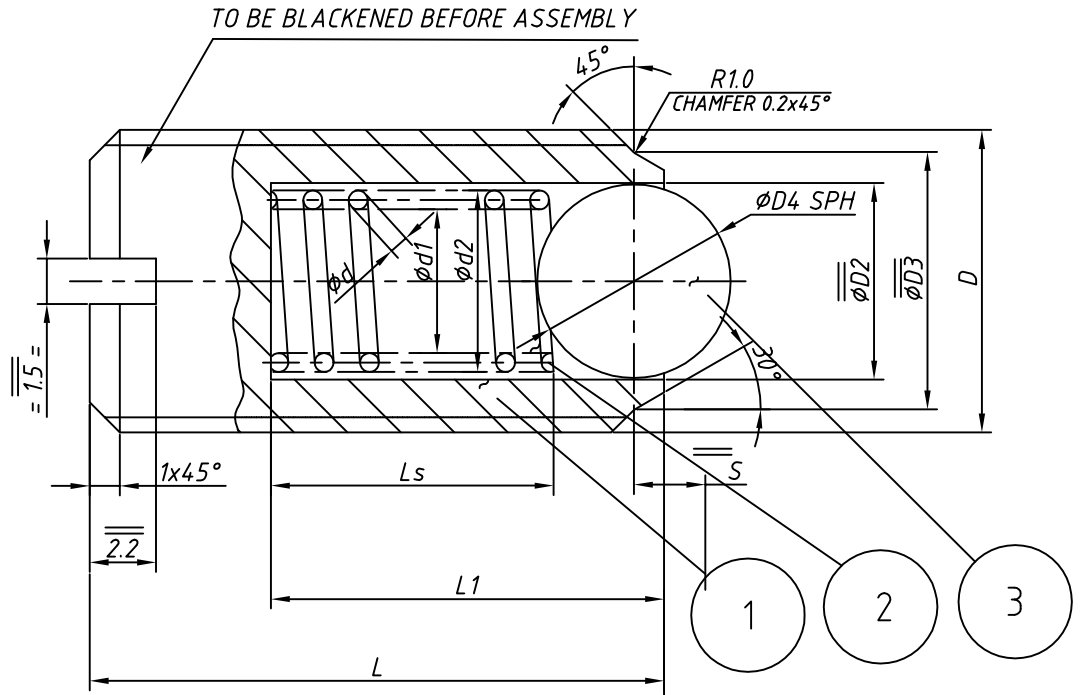
04-01-02-1,7-5

1/4"	6	CONNECTOR BODY	1	BRASS	HExA/F 24	41	
3/8"		CONNECTOR BODY	1	BRASS	HExA/F 30	50	
1/2"		CONNECTOR BODY	1	BRASS	HExA/F 36	65	
1/4"	5	CONNECTOR SLEEVE	1	BRASS	OD=27,ID=20.1	34	
3/8"		CONNECTOR SLEEVE	1	BRASS	OD=33,ID=24.1	43	
1/2"		CONNECTOR SLEEVE	1	BRASS	OD=40,ID=30.1	58	
1/4"	4	'O' RING	2		D=15.8,d=2.4		S I L T D NO=22
3/8"		'O' RING	2				S I L T D NO=23
1/2"		'O' RING	2		D=23.4,d=3.53		S I L T D NO=53
1/4"	3	G I PIPE	1		OD=13.2,ID=8.5		
3/8"		G I PIPE	1		OD=16.7,ID=10.5		
1/2"		G I PIPE	1		OD=21.0,ID=14.5		
1/4"	2	COPPER TUBE	1		OD=6.4,ID=4.0	TO BE SPECIFIED	
3/8"		COPPER TUBE	1		OD=9.5,ID=7.1		
1/2"		COPPER TUBE	1		OD=12.7,ID=10.3		
1/4"	1	HOSE CONNECTOR	2		d=Ø4,K=Ø9	30	
3/8"		HOSE CONNECTOR	2		d=Ø7,K=Ø12	30	
1/2"		HOSE CONNECTOR	2		d=Ø10,K=Ø15	30	
W	POS.	DESCRIPTION	QTY.	MAT.	FINISHED SIZE	LENGTH	REMARKS



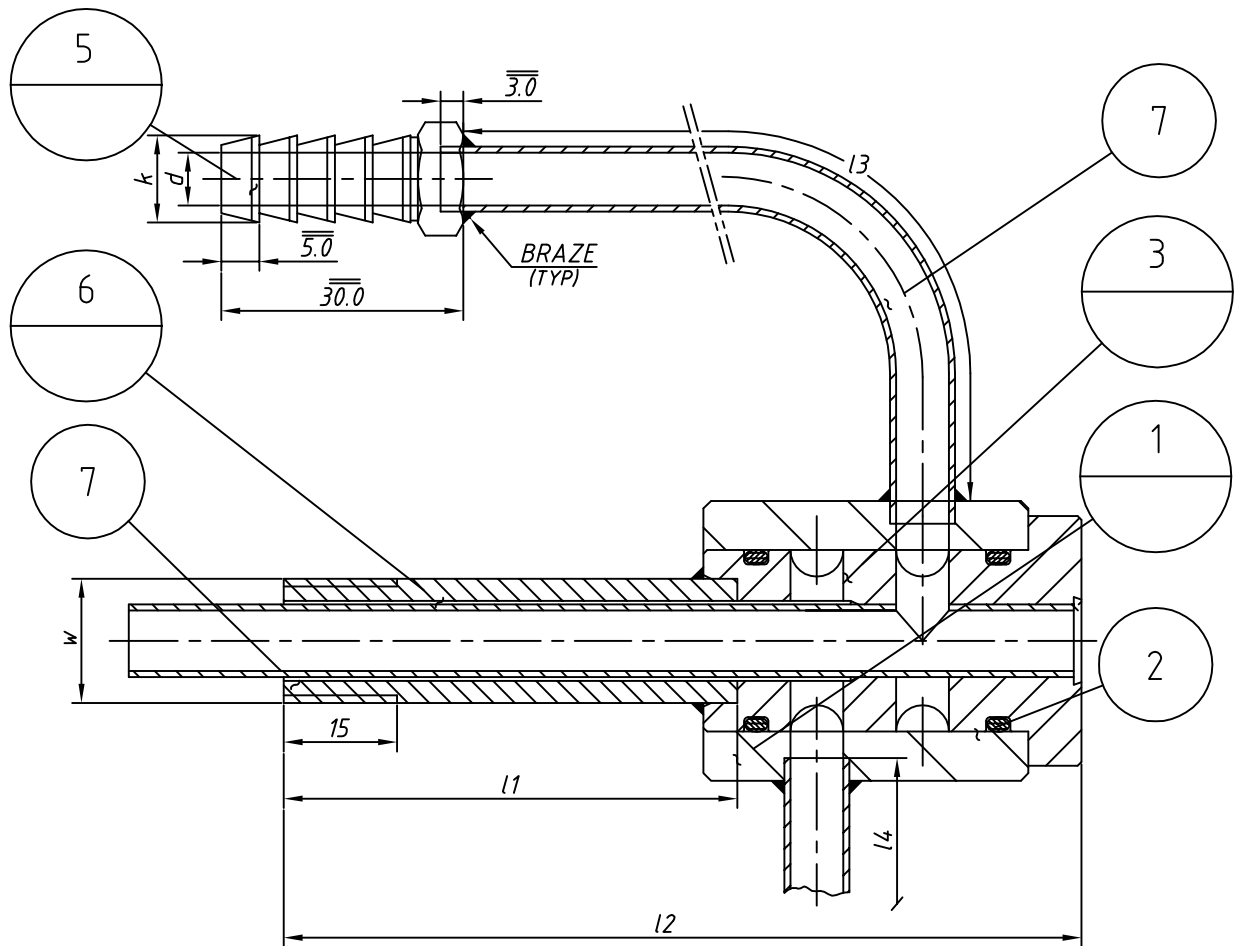
NO	d	D	$D1$	$D2$	Free L	Stroke Max	Load Max	Toatl No.of Coils		
1	1.5	7	8.5	10	10	2.7	11	4.25		
2	1.5	9	10.5	12	25	12.5	9	7.5		
3	1.5	11	12.5	14	50	32	9	11.3		
4	2	7	9	11	15	4	22	5		
5	2	9	11	13	35	14	19	10		
6	2	11	13	15	50	25	17	12		
7	2.5	9	11.5	14	25	7	32	6.8		
8	2.5	15	17.5	20	50	25	23	9.5		
9	3	9	12	15	15	26	48	4		
10	3	11	14	17	35	9.7	43	8.1		
11	3	13	16	19	50	17.5	40	10.5		
12	3.5	11	14.5	18	25	5.1	62	5.5		
13	3.5	15	18.5	22	50	17	52	9		
14	4	9	13	17	25	3	90	5.3		
15	4	11	15	19	50	9	80	10		
16	4	21	25	29	70	29	57	10		
17	4.5	15	19.5	24	35	7.4	94	6		
18	4.5	26	30.5	35	50	21.5	65	6		
19	5	11	16	21	25	2.6	135	4.3		
20	5	15	20	25	50	10	118	7.8		
21	5	21	26	31	80	25	100	10.5		
22	6	13	19	25	35	3.5	180	5		
23	6	26	32	38	70	20	130	8		
24	7	21	28	35	50	7.8	210	5.8		
25	7	26	33	40	35	6.6	182	3.9		
26	7	31	38	45	70	20	168	7		
27	8	26	34	42	50	8.7	260	5		
28	8	31	39	47	100	25	230	9		

DRN	01/10/09	SHAMNA MOHAN			
CHD	-	-			
APPR.	-	-			
NTTF BANGALORE			SPRINGS		04-01-01



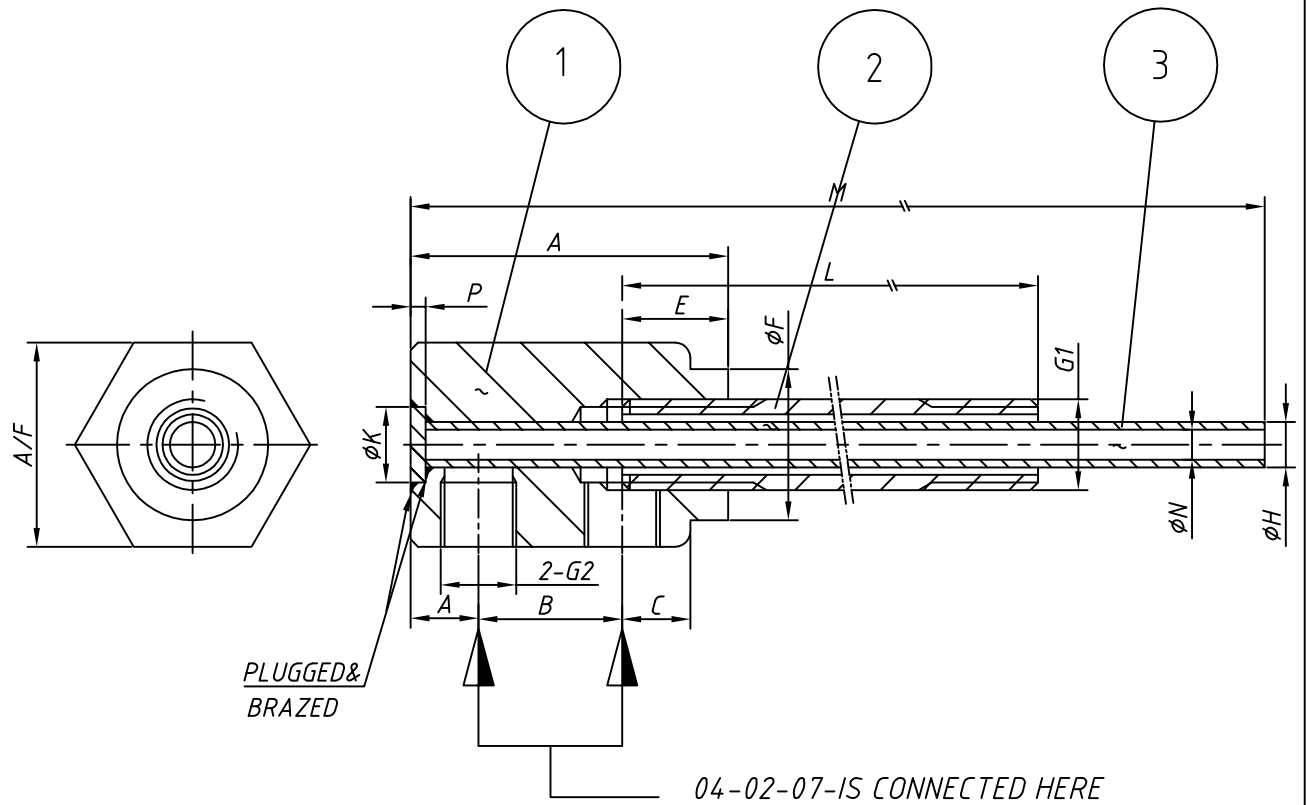
D	D2	D3	D4	d	d1	d2	L	L1	Ls	S
M10	6.05	7.5	6.4	0.6	4.3	5.5	19.0	13.0	13.0	1.0
M12	8.1	9.5	7.9	1.0	5.8	7.8	22.0	14.5	15.0	1.5

			3	BALL	1	SP. ST			
DRN	01/10/09	SHAMNA MOHAN	2	SPRING	1	SP. ST			
CHD	-	-	1	GRUB SCREW	1				
APPR.	-	-	POS.	DESCRIPTION	QTY.	MAT.	FINISHED SIZE	LENGTH	REMARKS
NTTF BANGALORE			SPRING STOPPERS				04-03-01		



TO SPECIFY-WRITE 04-02 05/W/l1,l2,l3,l4

DRN	01/10/09	SHAMNA MOHAN							
CHD	-	-							
APPR.	-	-	POS.	DESCRIPTION	QTY.	MAT.	FINISHED SIZE	LENGTH	REMARKS
NTTF BANGALORE			CASCADE WATER JUNCTION				04-02-05-1,3-1		



SL NO.	G1	AF	A	B	C	D	E	F	H	K	L	M	G2	N	P
1	1/4"	27	42	9	19	9	14	20	6	10	42	42	1/8"	4	2
2	3/8"	32	62	12	26	12	20	26	8	16	62	62	1/4"	6	3
3	1/2"	40	72	16	28	16	22	30	10	20	72	72	3/8"	8	4

TO SPECIFY WRITE 04-02-06/G1/M/L

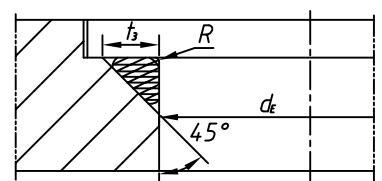
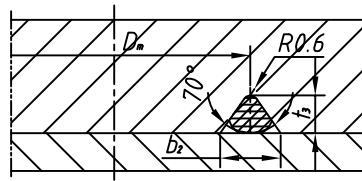
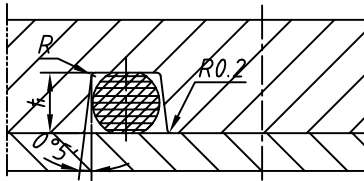
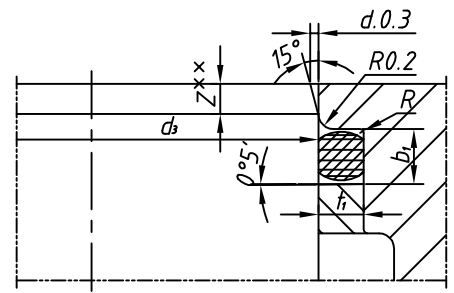
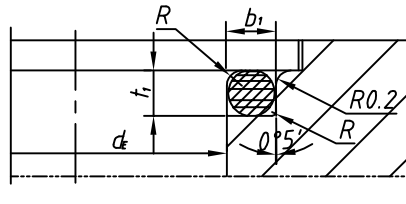
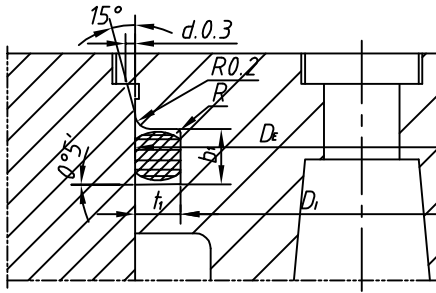
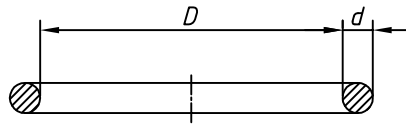
DRN	01/10/09	SHAMNA MOHAN	3	PIPE		MS			
CHD	-	-	2	G I PIPE					
APPR.	-	-	1	BODY		MS			
POS.	DESCRIPTION	QTY.	MAT.	FINISHED SIZE	LENGTH	REMARKS			
NTTF BANGALORE			CASCADE WATER JUNCTION				04-02-06		

D_e	D_1	d_e	D_A	D_m	b_1	b_2	R	t_1	t_2	t_3	D	d	<i>MFRS.REF.NO</i> ×		
													<i>Pioneer</i>	<i>S.I.NO</i>	
40	35	35	40	27.3	4.1	4.5	0.5	2.5	3	4.4	34.1	3.17		46	
25	20	20	25	22.3	4.1	4.5		2.5	3	4.4	19.1	3.18		47	
29	24	24	29	26.4							23.2			48	
35	30	30	35	31.8							28.6			49	
60	55	55	60	57.3							54.1			50	
17	11.6	11	16.4	14.4	4.4	4.9		2.7	3.2	4.7	11.0	3.4		51	
50	44.4	45	50.6	47.5	4.6	5		2.8	3.3	4.8	44	3.5		52	
25	19.4	19	24.6	22.23	4.6	5.1		2.8	3.3	4.8	18.7	3.53	<i>Pos3-125</i>		
27	21.4	21	26.6	23.8							20.22		<i>Pos3-126</i>		
28	22.4	22	27.6	25.6							21.82		<i>Pos3-129</i>		
30	24.4	24	29.6	26.9			23.4				<i>Pos3-131</i>		53		
36	30.4	30	35.6	33.3			29.79				<i>Pos3-137</i>		54		
38	32.4	32	37.6	34.9			31.34				<i>Pos3-138</i>		55		
42	36.4	36	41.6	39.1			35.52				<i>Pos3-140</i>				
43	37.4	37	42.6	39.6			36.1						56		
44	38.4	38	43.6	41.3			37.6				<i>Pos3-142</i>				
48	42.4	42	47.6	44.8			41.27				<i>Pos3-146</i>		57		
50	44.4	43	48.6	46.4			42.85				<i>Pos3-147</i>				
54	48.4	48	63.6	51.2			47.63				<i>Pos3-152</i>		58		
65	59.4	60	65.6	62.3			58.72				<i>Pos3-162</i>		59		
70	64.4	65	70.6	67.03			63.5				<i>Pos3-167</i>				
112	106.4	106	111.6	107.9			104.4						60		
140	134.4	135	140.6	136.4			132.9						61		
148	142.4	142	147.6	142.8	139.3		62								
155	149.4	150	155.6	149.1	145.6		63								
42	34.8	34	41.2	38	5.8	6.3	1	3.6	4.3	6.2	33.5	4.5		65	
48	39.4	38	46.6	42.8	6.9	7.4		4.3	5.1	7.3	37.5	5.33		66	
51	42.4	41	49.6	45.9							40.6			67	
60	51.4	51	59.6	55.4							50.03			68	
60	51.4	51	59.6	55.5							50.17		<i>Pos3-154</i>		
68	59.4	57	65.6	61.84							56.51		<i>Pos3-160</i>		
84	75.4	73	81.6	77.7							72.39		<i>Pos3-175</i>		
90	81.4	80	88.6	84.1							78.74		<i>Pos3-180</i>		
95	86.4	84	92.6	87.25							81.92		<i>Pos3-182</i>		
100	91.4	90	98.6	93.6							88.27		<i>Pos3-184</i>		

DRN	01/10/09	SHAMNA MOHAN			
CHD	-	-			
APPR.	-	-			
NTTF BANGALORE			O-RINGS		04-02-04-1,3-3

D_E	D_1	d_E	D_A	D_m	b_1	b_2	R	t_1	t_2	t_3	D	d	MFRS.REF.NO*		
													Pioneer	S.I.NO	
158	155.2	160	162.8	156.6	2.31	2.5	0.5	1.4	1.7	2.5	154.8	1.78		13	
18	14.8	15	18.2	16.0	2.6	3.0		1.6	1.9	2.7	14.0	2.00		14	
19	15.8	16	19.2	17.0							15.0			15	
21	17.8	18	21.2	19.0							17.0			16	
29	25.8	26	29.2	27.0							25.0			17	
14	10.4	10	13.6	12.2							2.9		3.3	1.8	2.1
12	8.4	8	11.6	9.9	3	3.4		1.8	2.2	3.2	7.6	2.3		19	
14	10.2	10	13.8	11.88	3.1	3.5		1.9	2.2	3.2	9.52	2.36		20	
15	11.2	11	14.8	12.7	3.1	3.6		1.9	2.3	3.3	10.3	2.40		21	
20	16.2	16	19.8	18.2							15.8			22	
24	20.2	20	23.8	22.2							19.8			23	
14	10	9.2	13.2	11.5							3.2		3.7	2	2.4
15	11	10	14	12.5	10			25							
20	16	16	20	17.5	15			26							
50	46	46	50	47.5	45.0			27							
20	15.8	16	20.2	18.14	3.4	3.8		2.1	2.5	3.6	15.54	2.60		28	
33	28.8	30	34.2	30.95							28.35			29	
14	9.8	10	14.2	11.75	3.4	3.84		2.1	2.5	3.6	9.13	2.62		30	
14	9.8	10	14.2	11.81							9.19		Pos3-111	31	
16	11.8	11	15.2	13.4							10.78		Pos3-113	32	
17	12.8	13	17.2	14.99			12.37				Pos3-116				
19	14.8	14	18.2	16.56			13.94				Pos3-118				
20	15.8	16	20.2	18.2			15.54				Pos3-120				
22	17.8	18	22.2	19.74			17.12				Pos3-122		33		
23	18.8	19	23.2	21.33			18.72				Pos3-124		35		
25	20.8	21	25.2	22.8			20.24				Pos3-127				
25	20.8	21	25.2	23.2			20.62				Pos3-128		36		
27	22.8	22	26.2	24.5			21.89						37		
28	23.8	24	28.2	26.4			23.79				Pos3-132		38		
38	33.8	35	39.2	37.2			34.59						39		
38	33.8	35	39.2	37.2			34.6						40		
17	12.2	12	16.8	14.5	3.9	4.4	2.4	2.9	4.1	11.5	3.00		41		
31	26.2	27	31.8	29.0						26.0			42		
75	70.2	70	74.8	72.5						69.5			43		
125	120.2	122	126.8	122.5						119.5			44		
80	75.2	76	80.8	78						75			45		

DRN	01/10/09	SHAMNA MOHAN	
CHD	-	-	
APPR.	-	-	
NTTF BANGALORE			0-RINGS
			04-02-04-1,3-2



D_E	D_1	d_E	D_A	D_m	b_1	b_2	R	t_1	t_2	t_3	D	d	$MFRS.REF.NO^{\times}$			
													Pioneer	S.I.NO		
8	5.6	5.2	7.6	6.5	1.9	2.4	0.5	1.4	1.7	2.5	5.0	1.5		1		
44	41.6	42	44.4	42.75							41.25			2		
27	24.4	24	26.6	25.39	2	2.5		1.3	1.5	2.2	23.79	1.6		3		
6	3.2	3	6.6	4.68	2.31			1.4	1.7	2.5	2.9	1.78				
8	5.2	5	7.8	6.25							4.47		Pos3-103	4		
8.5	5.7	5.5	8.3	7.06							5.28			5		
9.5	6.7	6.5	9.3	7.85							6.07		Pos3-105			
10	7.2	7	9.3	8.54							6.76		Pos3-106	6		
10	7.2	7	9.8	8.56							6.78			7		
11	8.2	8	10.8	9.42							7.64		Pos3-107			
12.5	9.7	9.5	12.3	11.02							9.24		Pos3-110	8		
14.2	11.4	11.2	14.0	12.88							11.1		Pos3-114			
16	13.2	13	15.8	14.2							12.42			9		
19	16.2	16	18.8	17.38							15.6			10		
35	32.2	32	34.8	33.28							31.5			11		
54	51.2	51	53.8	52.58							5.80			12		

DRN	01/10/09	SHAMNA MOHAN
CHD	-	-
APPR.	-	-

NTTF
BANGALORE

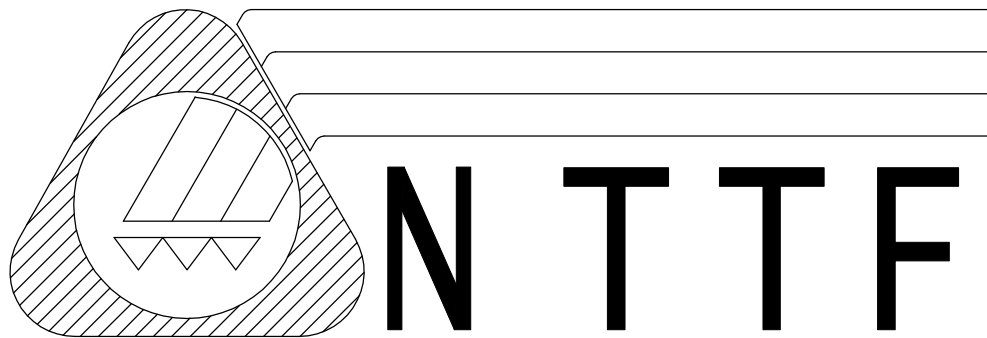
O-RINGS

04-02-04-1,3-1

PRESS

DESIGN

DETAILS



**COMPILED BY SRIVINAY
PGDTD 2009,
NTTF BANGALORE**

S.No	NTTF NUMBER SYSTEM	Company Name Press No	Capacity(tons)	Shut Ht(mm)	Stroke (mm)	Stroke/min	Ram Adjustment(mm)
1	H1	Godrej GE1-40-IG-51B	40	280	8-80	50	63
2	H2	Godrej GE1-40-I-C5	40	355	8-80	100	63
3	G4	Godrej FBI-63-S-1008/C	63	420	125	65	70
4	G2	Godrej GE1-63-IG-421-A	63	315	8-100	50	63
5		GodrejGE1 100 N16 FC 691	100	400	15-125	45	90
6	F1	HMT V-100OBPRE SG	100	400	8-125	40	80
7	F2	Godrej GE1-100-IGFC- 191/E	100	355	8-125	40	80
8	F4	Godrej EB1-100-S-1013	100	490	160	50	80
9		SEYI SN 1-100	110	350	180	35-65	90
10	E1	HMT P160 OBPR	160	500	160	35	125
11	E2	Godrej GE1-160-NIGFC- FB-784/C	160	425	25-160	40	100
12		SEYI SN 2-250	250	450	180	30-50	120

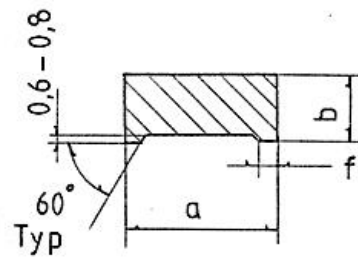
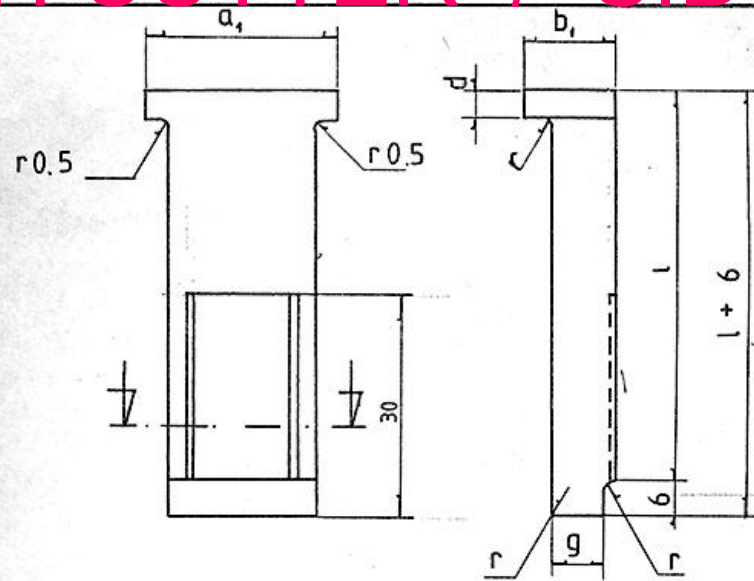
<u>PRESS TOOL MATERIALS:</u>			
COMPONENT	MATERIAL	COMPONENT	MATERIAL
1) Bottom pate	MS	8) Pilot	K100, D2=K110
2) Top plate	MS	9) End stopper	K720
3) Parallel block	MS	10) Finger stopper	K720, OHNS, EN31
4) Punch plate (Punch Holder)	MS	11) Box stripper	K720
5) Shank	MS (STD)	12) Traveling stripper	K720
6) Guide Pillar/	MS	13) Strip guides	K720
Guide Bush C-Clamps		14) Punch back plate	K720, EN8
Or L-Clamps		15) Guide pillar/Guide bush	E230
7) Die	K100, D2=K110	16) Spacer sleeve/Washer	E230
8) Punch	K100, D2=K110		

<u>PRESS DETAILS</u>		
PRESS	SHUT HEIGHT (mm)	STROKE (mm)
40 (1) S [52452]	200	75
40 (2) S	240	75
63 (1) S	225	60
63 (2)* S	225	60
63 (3) S	270	90
63 (4)	270	125
63 (5)	270	125
100 (1)* S [BRACKET II STAGE] [52450, 52457, 52697]	270	125
100 (2) S	270	100
100 (3)	270	160
100 (4)*	270	160
160 (1)* [52449, 52696]	270	160
160 (2)	270	125
315*	370	140

* - WITH DIE-CUSHION

<u>PLATE THICKNESS TO BE CONSIDERED</u>		
COMPONENT	MATERIAL	THICKENSS
TOP PLATE	MS	36,46,56,66
BACK PLATE	EN31	12 (HRC 52-56)
PUNCH HOLDER	EN8	22 (SOFT)
STRIPPER PLATE	EN8	22,32 (SOFT)
STRIPPER INSERT	EN31	16 (HRC 54-56)
DIE PLATE - AISI	D2	26,32,36,40 (HRC 58-60)
BOTTOM PLATE	MS	36,46,56,66
PARALLEL BLOCKS	MS	36,46,56,66
PIERCING PUNCH	AISI-D2	60,70,80 (HRC 58-60)
SHANK	EN8	(SOFT)
STRIP GUIDE	D2	(HRC 52-56)
SETTING BLOCK	EN8	(HRC 45-50) (SOFT)
GUIDE PILLAR	EN36	(HRC 60-62)
GUIDE BUSH	D2	(HRC 52-56)

PITCH CUTTER / SIDE CUTTER



$\alpha = \text{Pitch} + 0.05 \text{ mm}$		a_1	b	b_1	d	f	g	r
Till	6	$a + 6$	6	9	4	1.2	4	1
6	10	$a + 6$	6	9	4	1.6	4	1
10	16	$a + 6$	6	9	4	2.5	4	1
16	25	$a + 7$	8	11.5	5	3.0	5.5	1.5
25	40	$a + 8$	10	14.0	5	4.0	7.4	1.5
40	100	$a + 10$	12	16.0	5	5.0	9.0	1.5

DESIGNATION :

For $\alpha = 12 \text{ mm}$

$l = 60 \text{ mm}$

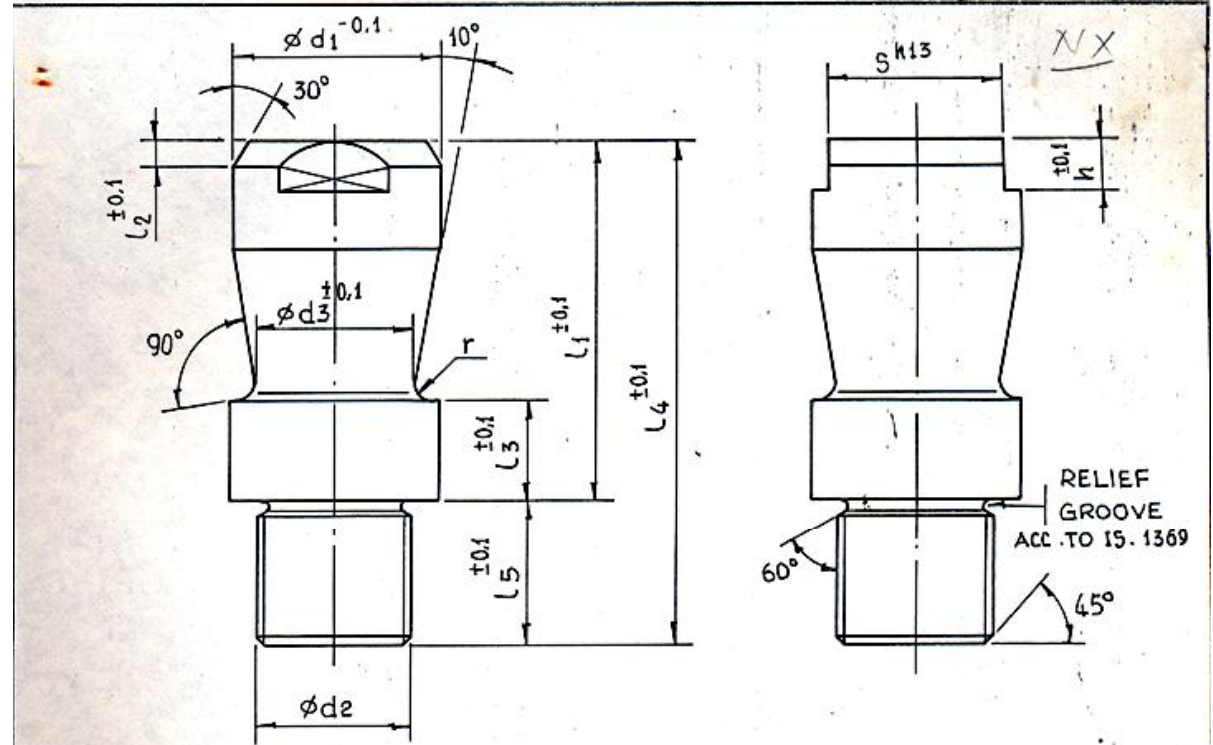
00001-20-01-1 / 12 / 60

MAT — HCHC

HARDNESS — TIP — 61 HRC

HEAD — 55 HRC

SHANK

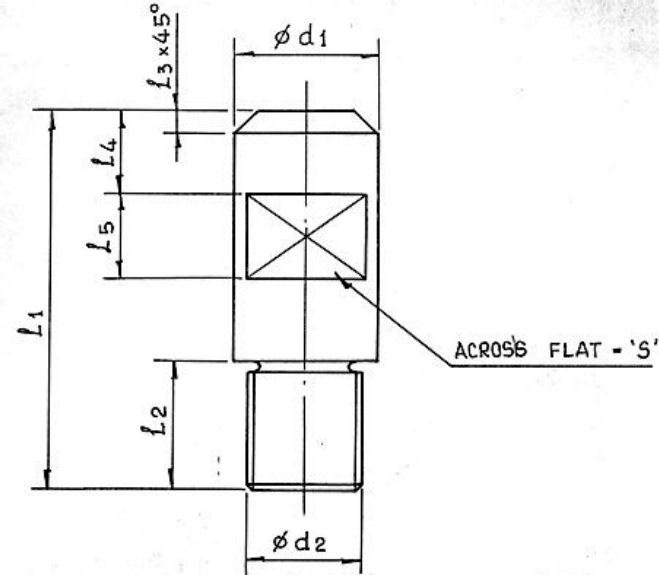


MATERIAL = DIN: 1.1730 OR AISI: 1040

INDEX	ϕd_1	ϕd_2	ϕd_3	L_1	L_2	L_3	L_4	L_5	h	r	S^{k13}
1	16	M10 x 1,25	12	32	2,5	10	47	15	6	2	13
2	20	M16 x 1,5	15	40	3	12	58	18	6	2,5	17
3	25	M16 x 1,5 M20 x 1,5	20	45	4	16	68	23	6	2,5	19
4	32	M20 x 1,5 M24 x 1,5	25	56	4	16	79	23	8	2,5	27
5	38	M24 x 1,5	32	60	5	26	83	23	10	4	32
6	40	M30 x 1,5		70			93				
7	50	M30 x 2	42	80	6	26	108	28	12	4	41
8		M36 x 2									
9	63	M42 x 3	53	100	8	26	128	28	16	4	55
10	80	M36 x 2	-	70	10	-	106	36	20	-	65
		M42 x 3									

TO ORDER SHANK OF DIA. $\phi d_1 = 40$, WRITE : 00003 20 / $\phi d_1 = 40$

DIN - 9859



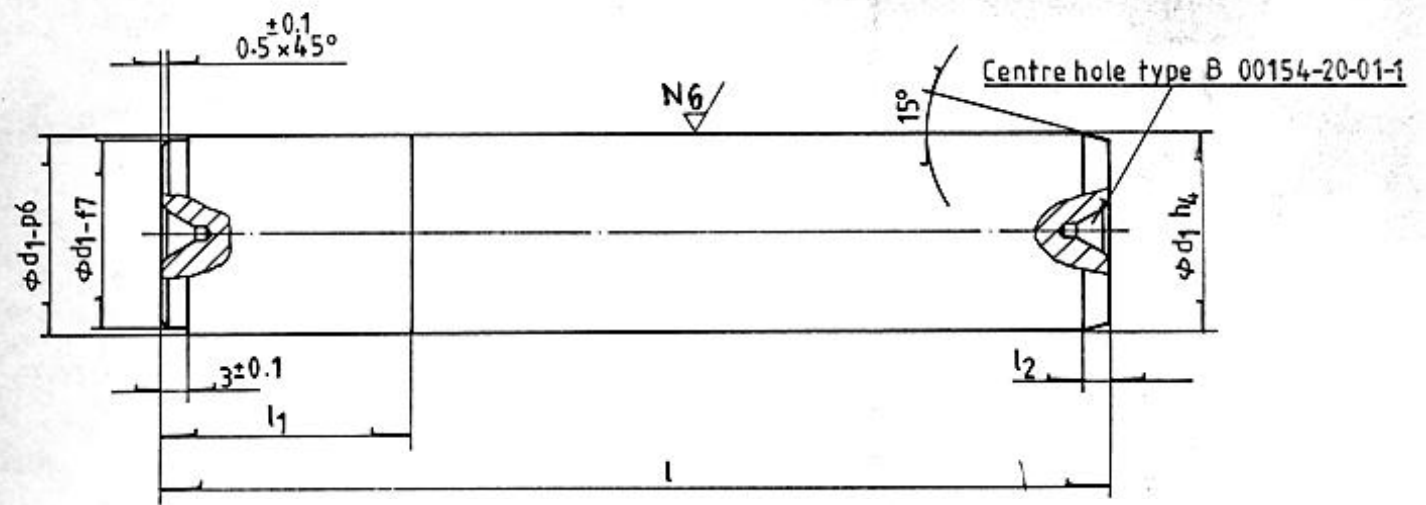
TOL. = ± 0.1

ϕd_1	ϕd_2	l_1	l_2	$l_3 \times 45^\circ$	l_4	l_5	'S'
20	M16 x 1,5	58	18	3	12	15,0	16
25	M16 x 1,5	68	23	4	15		21
25	M20 x 1,5	68	23	4	15		21
32	M20 x 1,5	79	23	4	20		27
32	M24 x 1,5	79	23	4	20		27
40	M24 x 1,5	93	23	5	27		36
40	M30 x 2	93	23	5	27		36
50	M30 x 2	108	28	6	32		46
65	M42 x 2	128	28	8	42		60

MATERIAL = DIN 1.1730 OR AISI = 1040

TO ORDER SHANK OF $\phi d_1 = 40$ WRITE = 00008 - 20 / $\phi d_1 = 40$

SHANK TYPE - 2

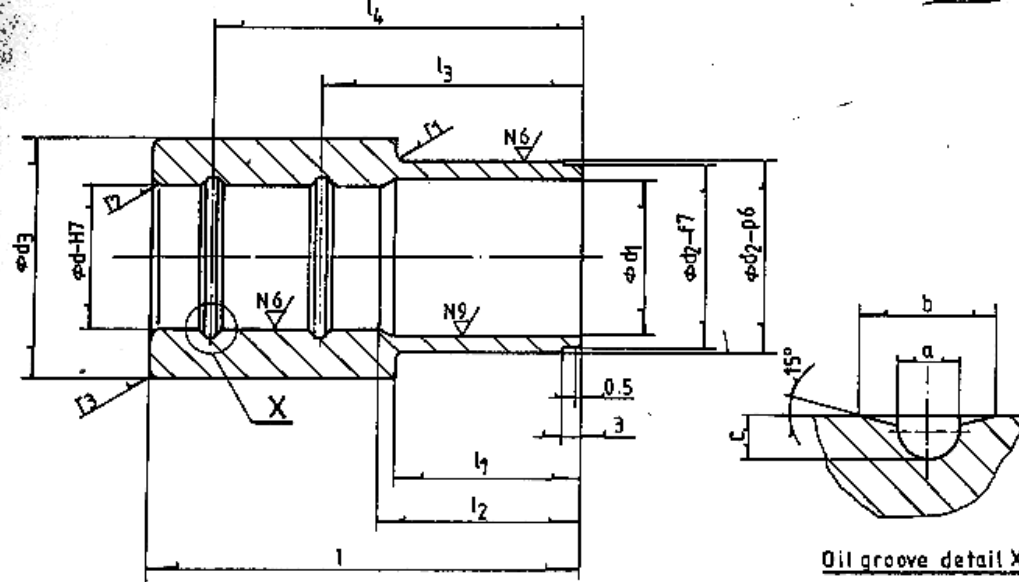


DIN: 1,5919 OR AISI : 3115 OR VEW: E230

CASE DEPTH=0.8

ϕd_1	$l_1^{+0.1}$	$l_2^{+0.1}$	$l^{+0.1}$										
10 / 12	15	3	80	100	120								
15 / 16	20	3	80	100	120	130	150	170					
						140	160	180					
18 / 19	25	3	80	100	120	130	150	170					
						140	160	180					
24 / 25	32	3		100	120	130	150	170	190	220			
						140	160	180	200				
30 / 32	40	5			120	130	150	170	190	220	240		
						140	160	180	200				
38 / 40	50	5				130	150	170	190	220	240	260	
						140	160	180	200				
48 / 50	63	5					150	170	190	220	240	260	280
							160	180	200				
60 / 63	80	5					150	170	190	220	240	260	280
							160	180	200				300

GUIDE PILLAR



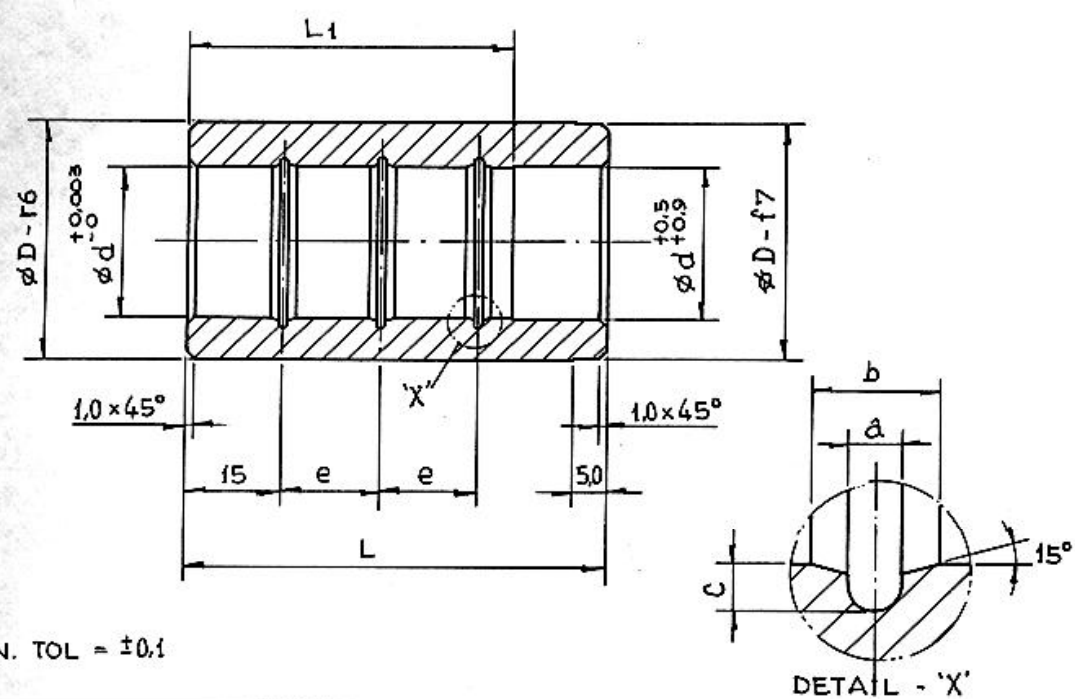
To ± 0.1

$\phi d-H7$	ϕd_1	ϕd_2	ϕd_3	l	l_1	l_2	l_3	l_4	r_1	r_2	r_3	a	b	c
10/12	12.5	18	22	40	18	19	24	34	0.8	1	1	1	2	1
15/16	16.5	22	25	50	20	22	30	42	1	1.5	1.5	1.2	3	1.2
18/19	20	25	32	60	22	25	32	50	1.5	2	2	1.6	4	1.5
24/25	26	32	40	70	28	32	42	60	1.5	2.5	2	2	4.5	1.6
30/32	34	40	50	85	36	40	52	74	1.5	2.5	2	2.5	5	1.8
38/40	42	50	63	110	46	50	65	95	1.5	3	2.5	2.5	6	2
48/50	52	63	73	140	58	63	82	120	2	3	2.5	2.8	6.5	2.5
60/63	65	78	88	175	75	80	105	150	2	3	2.5	3	7	2.8

To order a bush of dia 24/25 Write : 00005-20/24/25

Mat: DIN=1.5919 OR AISI=3115 OR VEW=E230
HON: CASE DEPTH=0.8

GUIDE BUSH



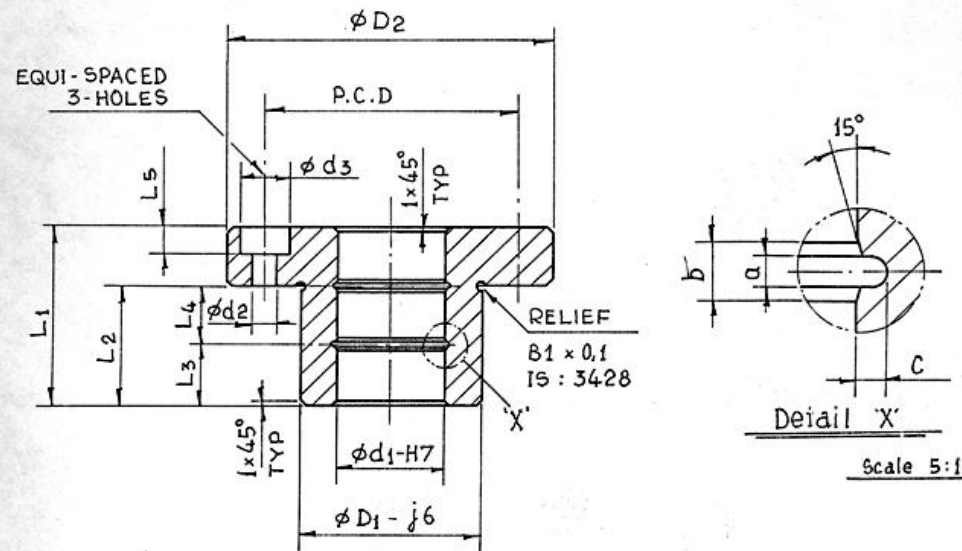
ϕd	ϕD	L										L ₁	a	b	c	e
		25	35	50	65	80	100	120	140	160						
10 / 12	20											24	0.8		0.8	15
15 / 16	25											32	1	2	1	15
18 / 19	30											40	1	2	1	15
24 / 25	38											50	1.6	4	1.5	15
30 / 32	45											64	1.6	4	1.5	20
38 / 40	56											80	2	5	1.7	20
48 / 50	73											100	2	5	1.7	20
60 / 63	85											120	2.5	6	2	25

TO ORDER A BUSH OF $\phi d = 25$; $L = 65$ WRITE : 00006 - 20 / 25 / 65

CASE DEPTH = 0.8

MATERIAL = DIN 1.5919
 AISI = 3115
 VEW = E 230

SLEEVE GUIDE BUSH



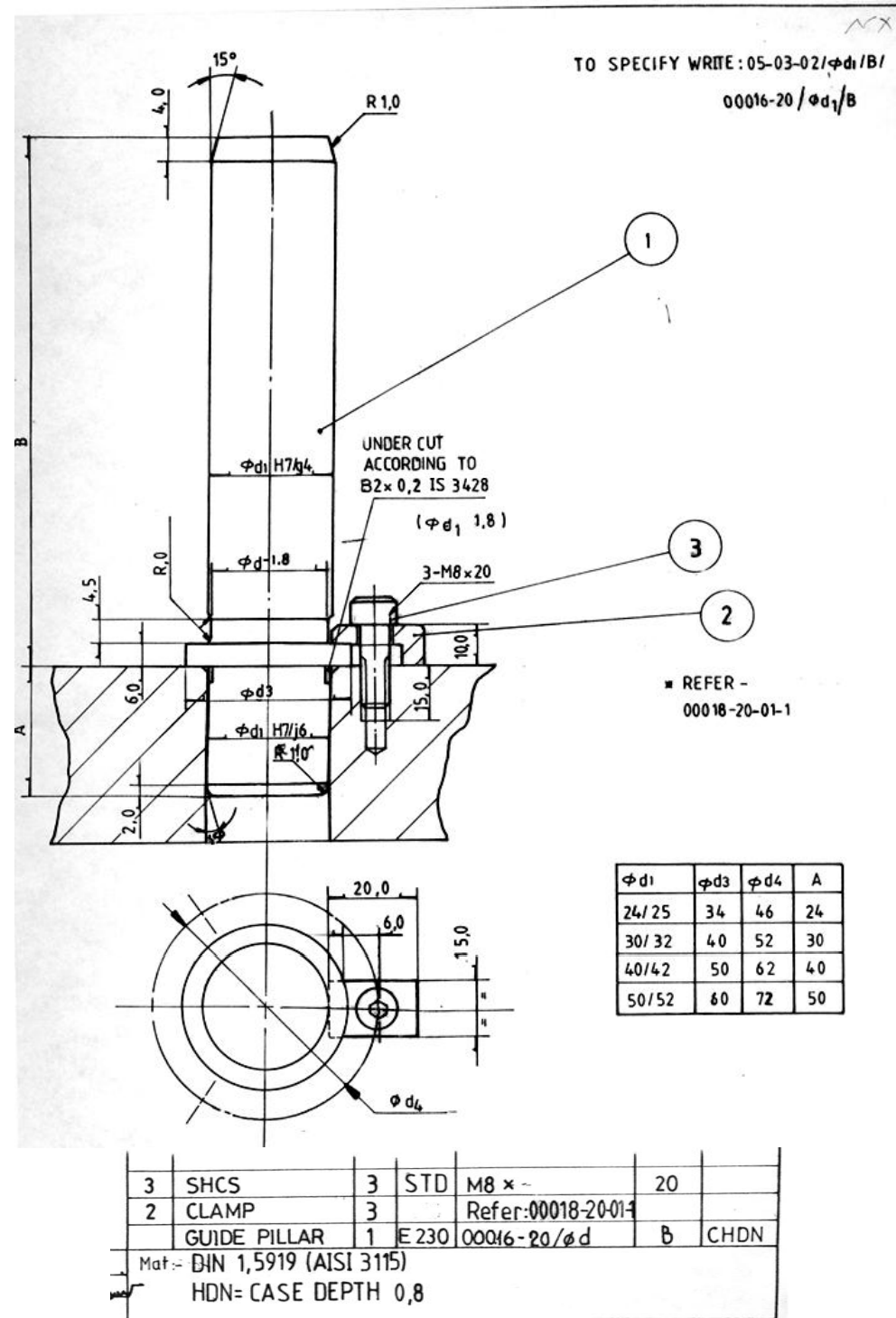
TOL = ± 0.1

$\phi d_1 - H7$	ϕD_1	ϕD_2	P.C.D	L_1	L_2	L_3	L_4	ϕd_2	ϕd_3	L_5	a	b	c	SCREW
10 / 12	20	42	30	30	20	10	10	4,2	8,5	4,5	1	2	1	M4
15 / 16	25	50	35	30	20	10	10	4,2	8,5	4,5	1	2	1	M4
18 / 19	30	54	42	30	20	10	10	4,2	8,5	4,5	1	2	1	M4
24 / 25	35	60	48	38	25	12	14	4,2	8,5	4,5	1,6	4	1,5	M4
30 / 32	42	68	56	42	30	12	18	4,2	8,5	4,5	1,6	4	1,5	M4
38 / 40	52	80	66	52	40	15	22	5,2	10,5	5,5	2	5	1,7	M5
48 / 50	60	88	74	58	45	15	28	5,2	10,5	5,5	2	5	1,7	M5
60 / 63	75	104	90	62	50	15	32	5,2	10,5	5,5	2	5	1,7	M5

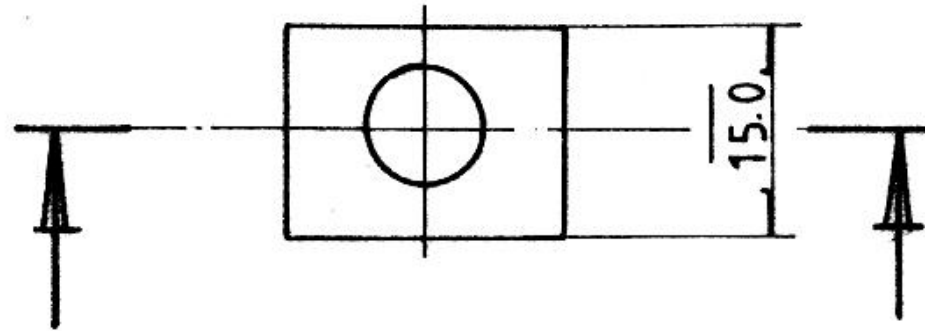
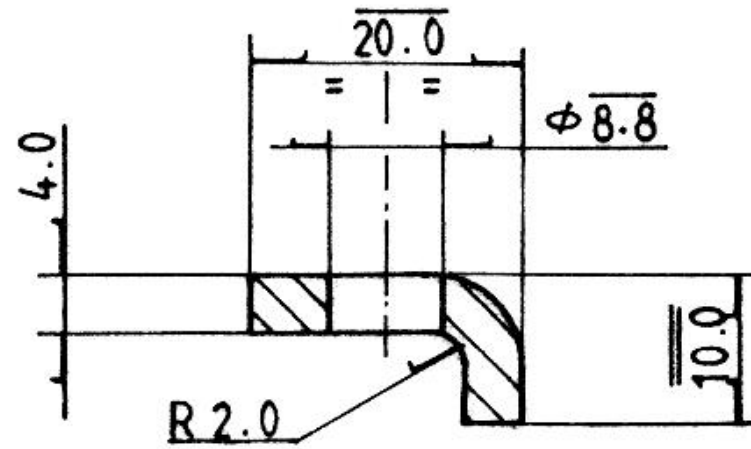
TO ORDER STRIPPER BUSH OF $\phi d_1 = 16$ WRITE 00007 20/ $\phi d_1 = 16$.

MATL. = DIN 1.5919 OR AISI: 3115 OR VEW: E230
HDN = CASE DEPTH 0,8

STRIPPER PLATE GUIDE BUSH

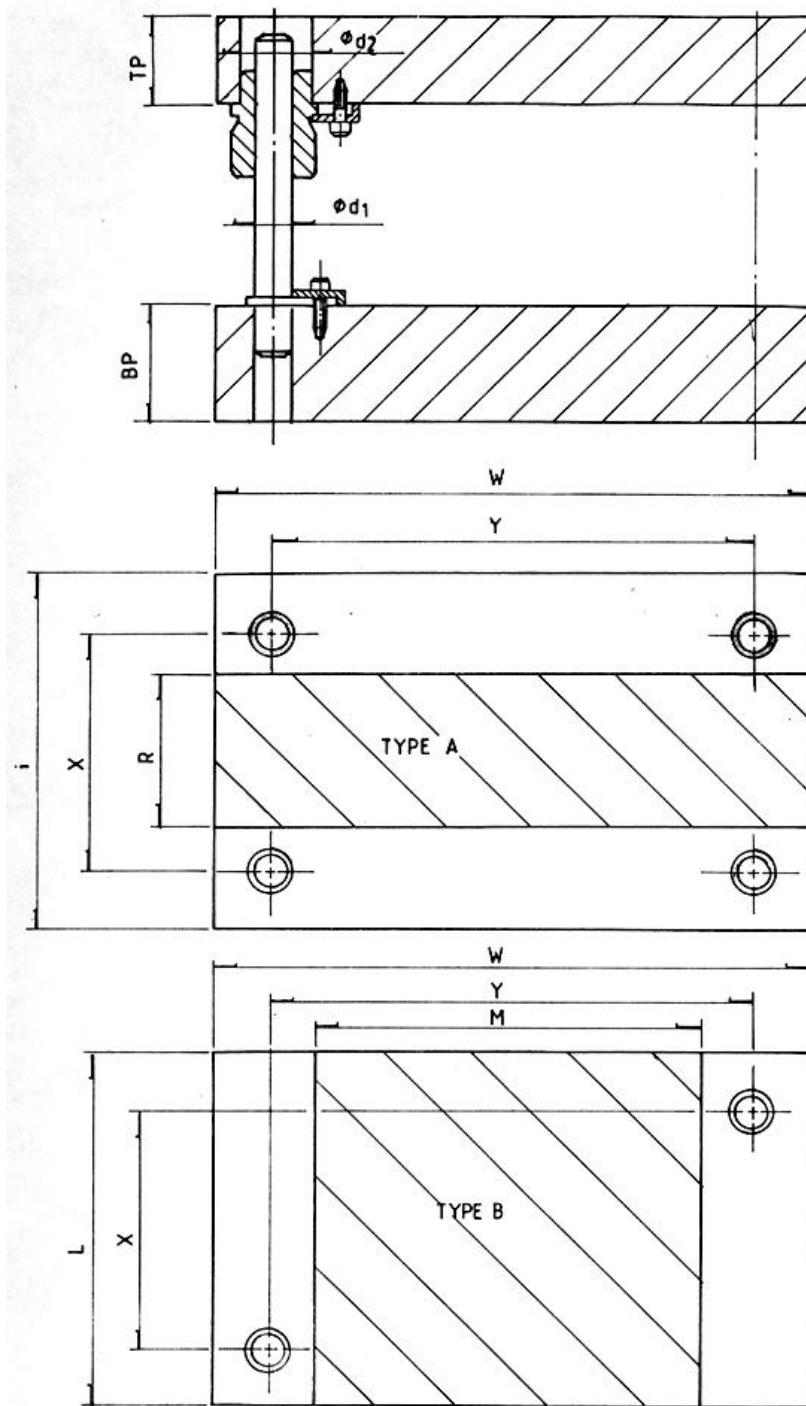


DEMOUNTABLE GUIDE PILLAR



MATERIAL : AISI – 1040

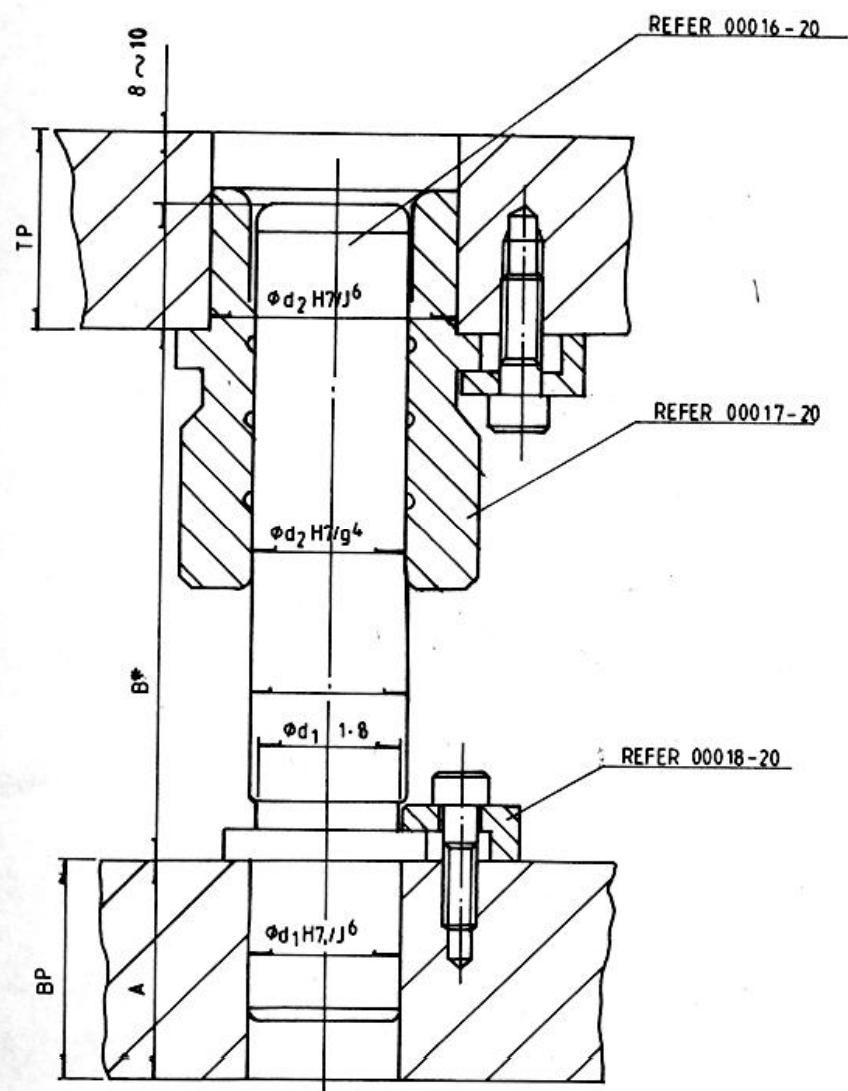
CLAMP



STANDARD DIE SETS

SERIES	L	W	X	Y	TP	BP	TYPE A		TYPE B		ϕd_1	ϕd_2
							R	W	M	L		
2020	196	196	120	120	36/46	36/46	50	196	50	196	24/25	42
2025	196	246	100	150	36/46	46/46	40	246	90	196	30/32	50
2030	196	296	100	200	36/46	46/46	40	296	140	196	30/32	50
2035	196	346	100	250	46/56	56/56	40	346	190	196	30/32	50
3030	296	296	200	200	46/56/66	46/56/66	140	296	140	296	30/32	50
3035	296	346	200	250	46/56/66	46/56/66	140	346	190	296	30/32	50
3035	296	346	172	222	46/56/66	46/56/66	91	346	141	296	50/52	72
3040	296	396	186	286	56/66	56/66	117	396	217	296	40/42	60
3040	296	396	172	272	56/66	56/66	91	396	191	296	50/52	72
3045	296	446	186	336	56/66	56/66	117	446	267	296	40/42	60
3045	296	446	172	372	56/66	56/66	91	446	241	296	50/52	72
3050	296	496	172	372	56/66	56/66	91	496	291	296	50/52	72
4040	396	396	286	286	56/66	56/66	217	396	217	396	40/42	60
4040	396	396	272	272	56/66	56/66	191	396	191	396	50/52	72
4045	396	446	272	322	56/66	56/66	191	446	241	396	50/52	72
4050	396	496	272	372	56/66	56/66	191	496	291	396	50/52	72
4060	396	596	272	472	56/66	56/66	191	596	391	396	50/52	72

STANDARD DIE SETS

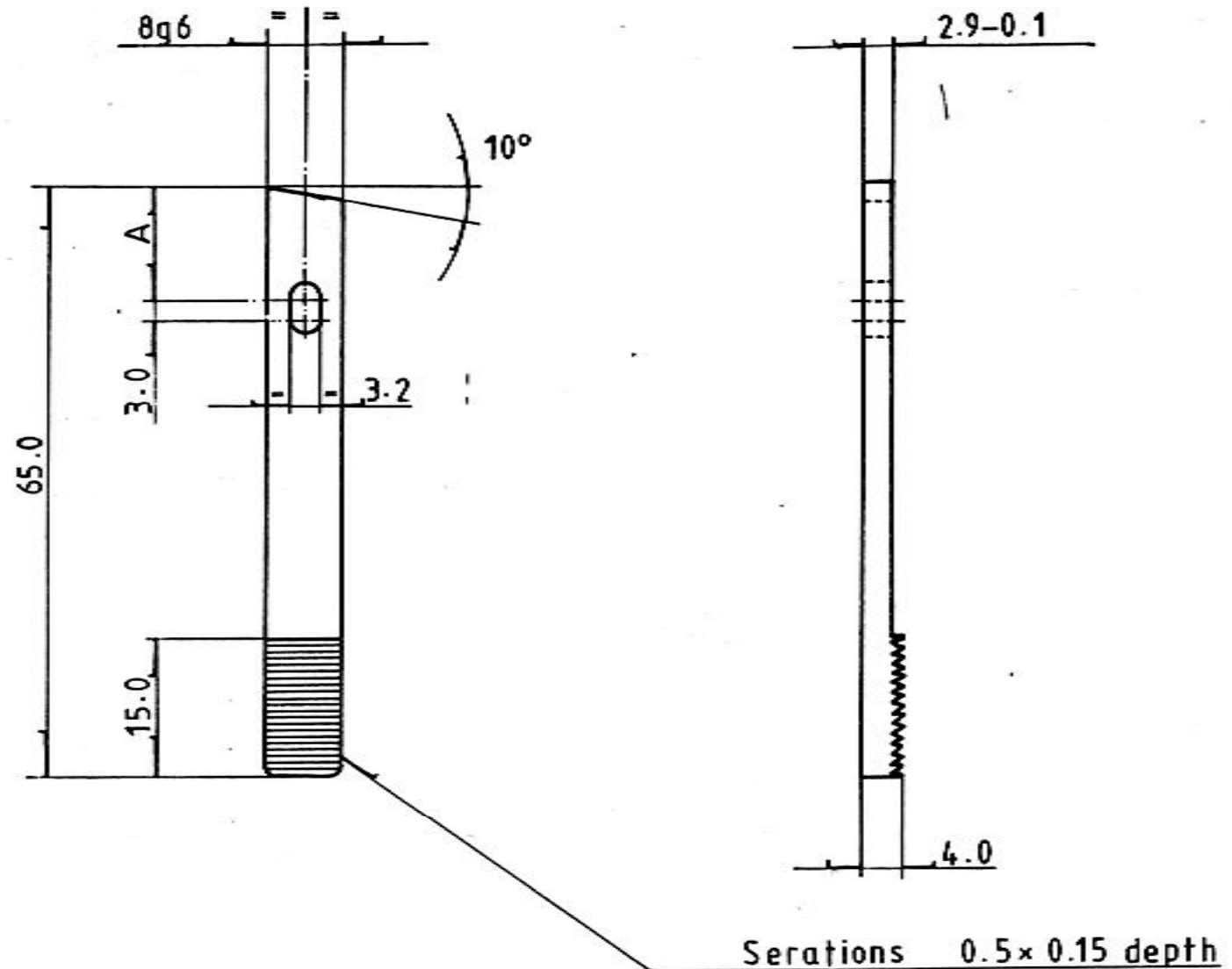


ϕd_1	H7		g4		H7/g4		j6		H7/j6	
ϕd_1	er	es	er	es	MIN CL	MAX CL	er	es	MAX INT	MAX CL
24/25	0	+21	-13	-7	7	34	-4	+9	9	25
30/32	0	+25	-16	-9	9	41	-5	+11	11	30
40/42	0	+25	-16	-9	9	41	-5	+11	11	30
50/52	0	+30	-18	-10	10	48	-7	+12	12	37

NOTE : TNT - INTERFERENCE
CL - CLEARANCE

STANDARD DIE SETS

FINGER STOP

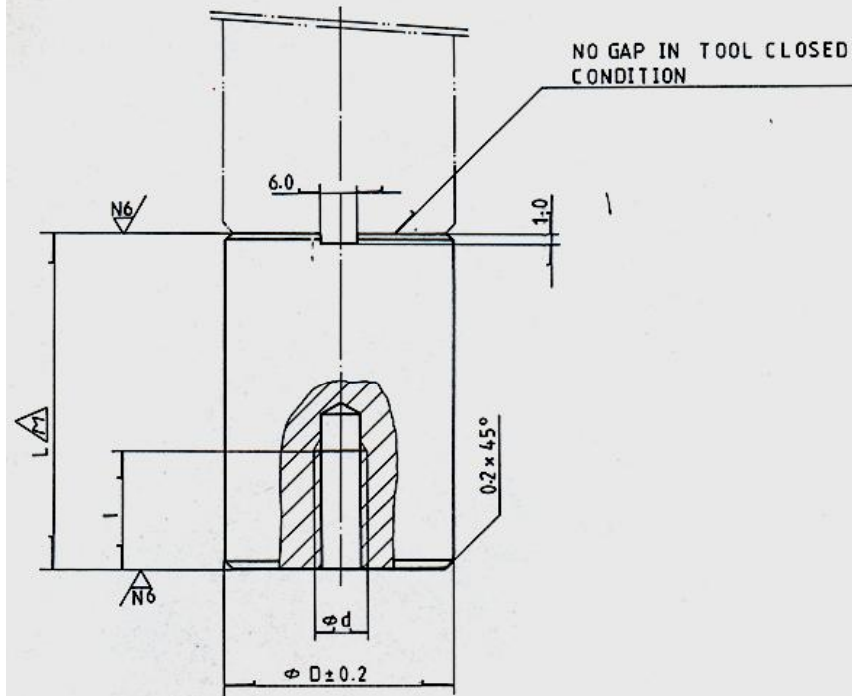


To specify write 00035-20/A

MAT : AISI D2

HARDNESS HRC = 55-58

SETTING BLOCK -1



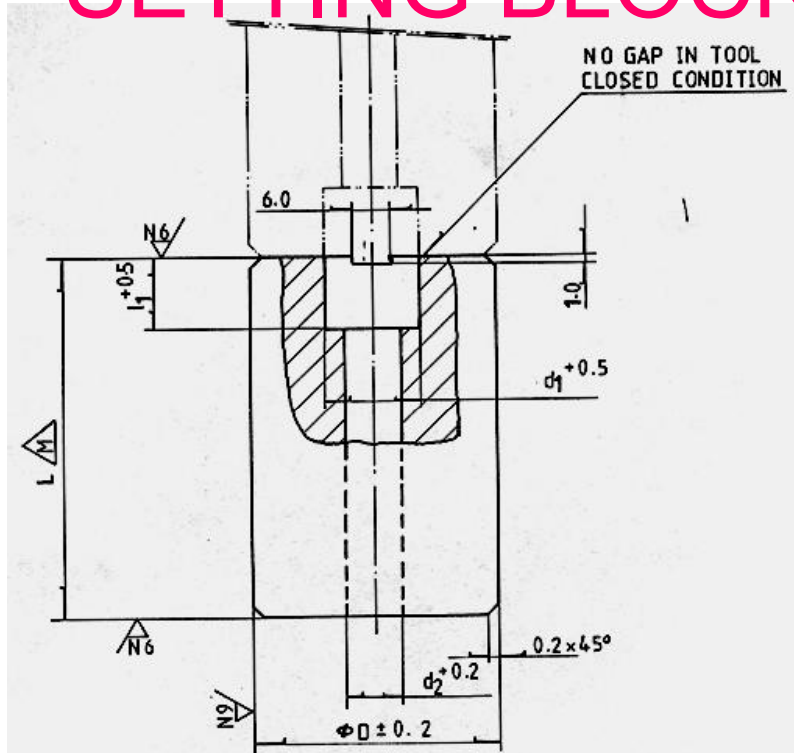
ϕD	ϕd	l	L
15 - 25	M6	15	
25 - 45	M8	20	
ABOVE 45	M10	25	

TO SPECIFY WRITE : 00034 - 20 / ϕD / L

MATERIAL : AISI - D2

HARDNESS HRC 55 - 58

SETTING BLOCK -2



ϕD	d_1	d_2	l_1	L
20 - 30	11.0	7.0	7.0	
30 - 45	14.0	9.0	9.0	
ABOVE 45	17.0	11.0	11.0	

To specify write 00033-20/ ϕD /L

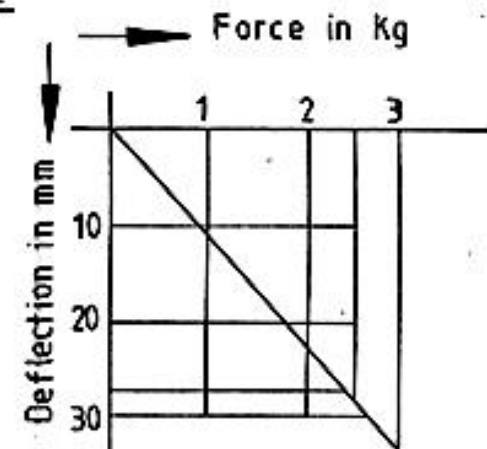
MAT : AISI D2

HARDNESS : HRC 55-58

PCD 36.00 M4 3 N2 S
EQUISPACED



Wire $\phi = 1\text{ mm}$
 Inside $\phi = 13\text{ mm}$
 Free length = 36.3 mm
 No of coils = 4
 Max stroke = 28 mm
 Max force = 2.5 kgs



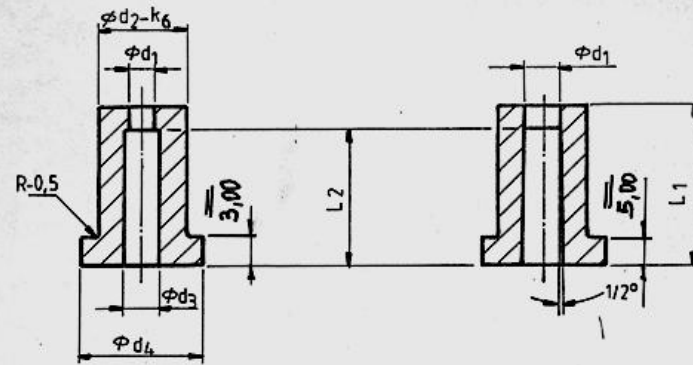
RUN STOPPER

TO ORDER RUN STOPPER WITH PIN DIA 'D' = 12

SPECIFY = 00039-20 / D-12

6	ALLEN SCREW	3	STD	M4 x	15	
5	SPRING	1	STD	ϕ 1 x	SUIT	
4	BUSH	1	D2	ϕ 24 x	16.5	HRC= 56
3	PIN	1	D2	ϕ 16 x	51	HRC= 56
2	HANDLE	1	D2	4	60	
1	SLEEVE	1	1040	48 x	25	
POS	DESCRIPTION	ITEM	MAT	FINISH SIZE	LENGTH	REMARKS

DIE BUTTON



Tol = $\pm 0,1$

TYPE 'A'

TYPE 'B'

TYPE	ϕd_1		$\phi d_2 k_6$	ϕd_3	ϕd_4	L_1	L_2
A	1,00	1,80	6,00	$\phi d_1 + 1,5$	12,0		
	1,81	2,60	10,00		16,0		
	2,61	3,30	12,00		18,0		
	3,31	4,00	15,00		21,0		
B	4,01	5,00	15,00		21,0		
	5,01	6,00	18,00		24,0		
	6,01	8,00	22,00		30,0		
	8,01	10,00	25,00		32,0		
	10,01	12,00	30,00		38,0		
	12,01	15,00	32,00		40,0		
	15,01	18,00	36,00		44,0		
	18,01	22,00	40,00		48,0		
	22,01	25,00	45,00		54,0		

To specify write = 0041-20- $\phi d_1/L_1/L_2$

MATL: AISI-D2 (DIN 1,2379)

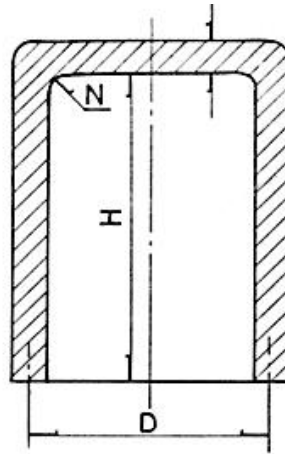
HARDNESS = HRC 60-62

SHEAR STRENGTH - MAX / USS

			USS-ULTIMATE SHEAR STRENGTH
Material	Max		Remarks
Steel with 0.1% Carbon	24 — 30	Kg/mm ²	Deep drawing steel
" 0.2% Carbon	32 — 40	Kg/mm ²	
" 0.3% Carbon	36 — 48	Kg/mm ²	
" 0.4% Carbon	45 — 56	Kg/mm ²	
" 0.6% Carbon	55 — 70	Kg/mm ²	
" 0.8% Carbon	70 — 90	Kg/mm ²	Spring Steel
Si. Steel	45 — 55	Kg/mm ²	
Stainless Steel	30 — 40	Kg/mm ²	
Copper	20 — 30	Kg/mm ²	
Brass 60	35 — 40	Kg/mm ²	
Bronze	36 — 45	Kg/mm ²	
German Silver	30 — 35	Kg/mm ²	Cu: 45 — 70% Zn 12 — 48% Ni: 8 — 20%
Tin	3 — 4	Kg/mm ²	
Zinc	10 — 12	Kg/mm ²	
Lead	2 — 3	Kg/mm ²	
Pure Aluminium	8 — 12	Kg/mm ²	99.5%
Al. Magnesium Alloy	32 — 75	Kg/mm ²	
Al. Silicon Alloy	38 — 78	Kg/mm ²	
Paper or Cardboard	2 — 5	Kg/mm ²	
Hard Board	7 — 9	Kg/mm ²	
Laminated paper (Formica)	10 — 14	Kg/mm ²	
Resin impregnated paper			
Laminated fabric or	9 — 12	Kg/mm ²	
Resin impregnated fabric			
Mica	5 — 8	Kg/mm ²	
Plywood	2 — 4	Kg/mm ²	
Celluloid	4 — 6	Kg/mm ²	
Leather	0.7	Kg/mm ²	
Soft Rubber	0.7	Kg/mm ²	
Hard Rubber	2 — 6	Kg/mm ²	

DRAWS

NUMBER OF DRAWS



D = MEAN DIAMETER

H = INSIDE HEIGHT

$N \geq 4T$

T = SHEET THICKNESS

P = PROBABLE NUMBER
OF REDUCTIONS OR DRAWS

SHALLOW DRAW - A drawn piece part which requires one drawing operation is considered to be a shallow draw.

DEEP DRAW - A drawn piece part which requires more than one drawing is considered to be a deep draw.

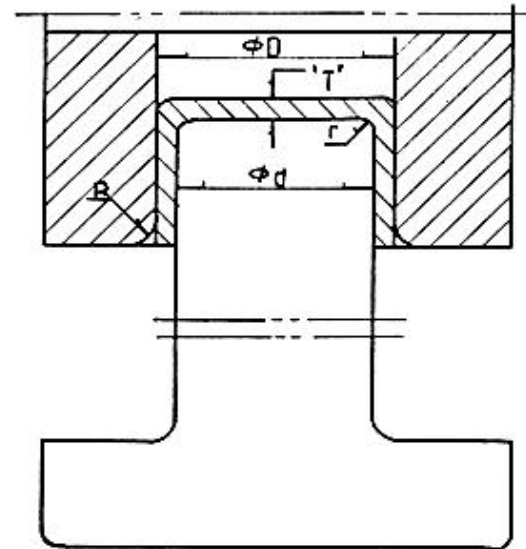
DRAW	$\frac{H}{D}$ RATIO
SHALLOW	5/8 to 3/4
DEEP	> 3/4

Nº OF DRAWS - Probable number of reductions or draws will depend on $\frac{H}{D}$ ratio.

$H \div D$	P
< 0.75	1
0.75 - 1.5	2
1.5 - 3.0	3
3.0 - 4.7	4

DRAW CLEARANCE

RECOMENDED PRACTICAL CLEARANCE IN DRAWING



DRAW CLEARANCE

$$= \frac{D-d}{2}$$

T = Thickness of sheet

ϕd = Draw punch dia

ϕD = Draw DIE dia

R = Draw Radius

T - mm	FIRST DRAW	RE - DRAW	SIZING DRAW
< 0.40	1.08T	1.09T - 1.10T	1.04T - 1.05T
0.40-1.3	1.08T - 1.10T	1.10T - 1.13T	1.05T - 1.06T
1.30-3.0	1.10T - 1.13T	1.13T - 1.15T	1.06T - 1.08T
> 3.0	1.13T - 1.15T	1.15T - 1.20T	1.08T - 1.10T

DRAW CLEARANCE

It is the gap between the drawing die and the draw punch which controls the thickness of the sheet.

DRAW REDUCTION PERCENTAGE FOR STEEL

PRACTICAL REDUCTION PERCENTAGE FOR DRAWING & RE DRAWING

MATERIAL = STEEL - DRAW QUALITY OR BETTER

STOCK MATERIAL THICKNESS mm	% 1 st DRAW	% 2 nd DRAW	% 3 rd DRAW	% 4 th DRAW
< 1.6	40-48	20 - 25	18-20	16 - 18
1.6 - 3.2	40-48	15 - 18	14-15	13 - 14
3.2 - 4.7	40-48	12 - 15	11-12	10 - 11
4.7 - 6.3	40-48	10 - 13	9-10	8-9
> 6.3	40-48	8 - 10	7-8	6-7

DRAW REDUCTION PERCENTAGE FOR BRASS

MATERIAL = BRASS DRAWING QUALITY

STOCK MATERIAL THICKNESS mm	% st DRAW	% nd DRAW	% rd DRAW	% th DRAW
< 1.6	44 - 50	20 - 29	18 - 23	18 - 20
1.6 - 3.2	44 - 50	16 - 23	15 - 17	12 - 15
3.2 - 4.7	44 - 50	14 - 17	12 - 14	11 - 13
4.7 - 6.3	44 - 50	10 - 14	9 - 11	8 - 10
> 6.3	44 - 50	9 - 11	8 - 9	7 - 8

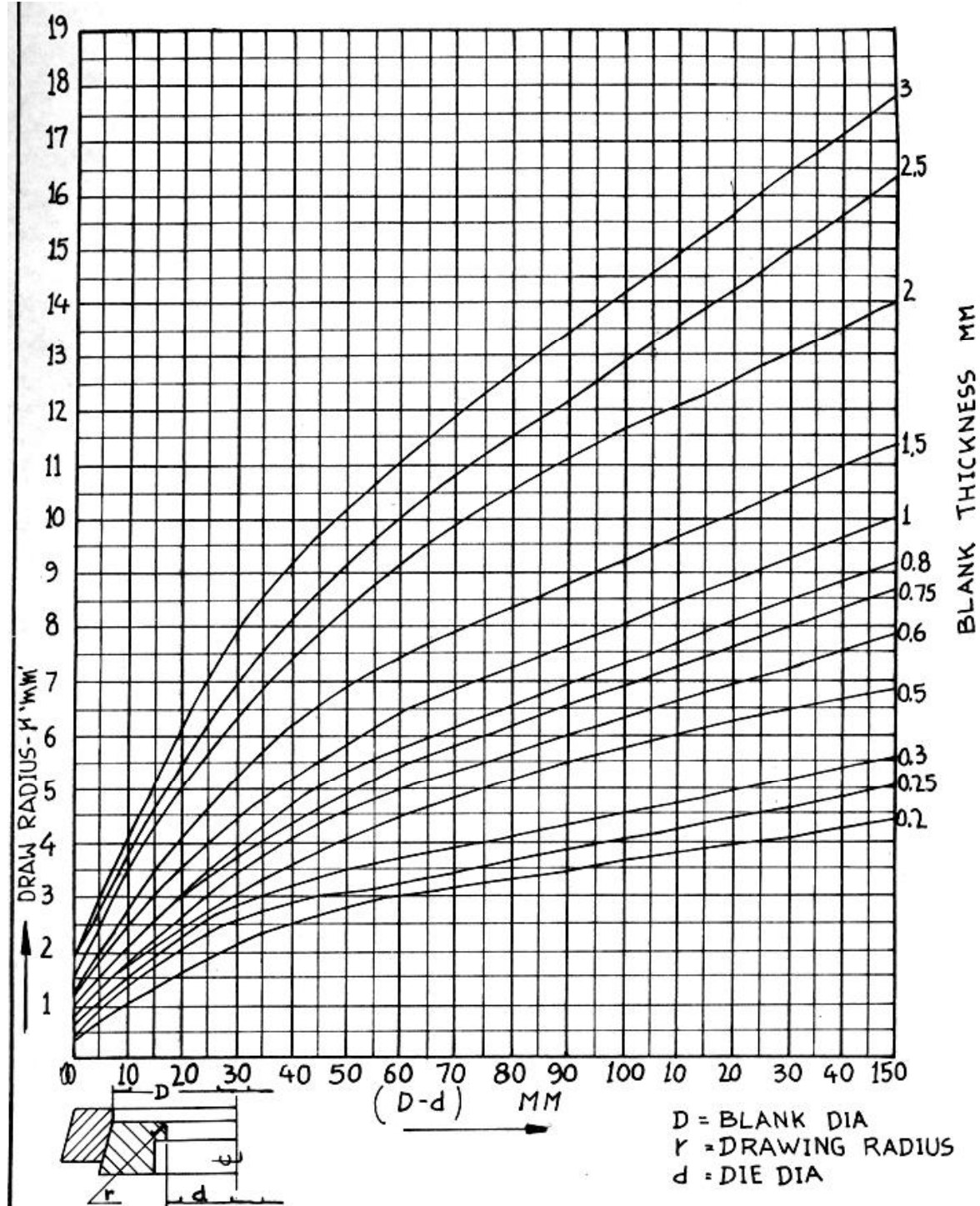
DRAW REDUCTION RATIO FOR ALUMINIUM

PRACTICAL DRAW REDUCTION RATIO

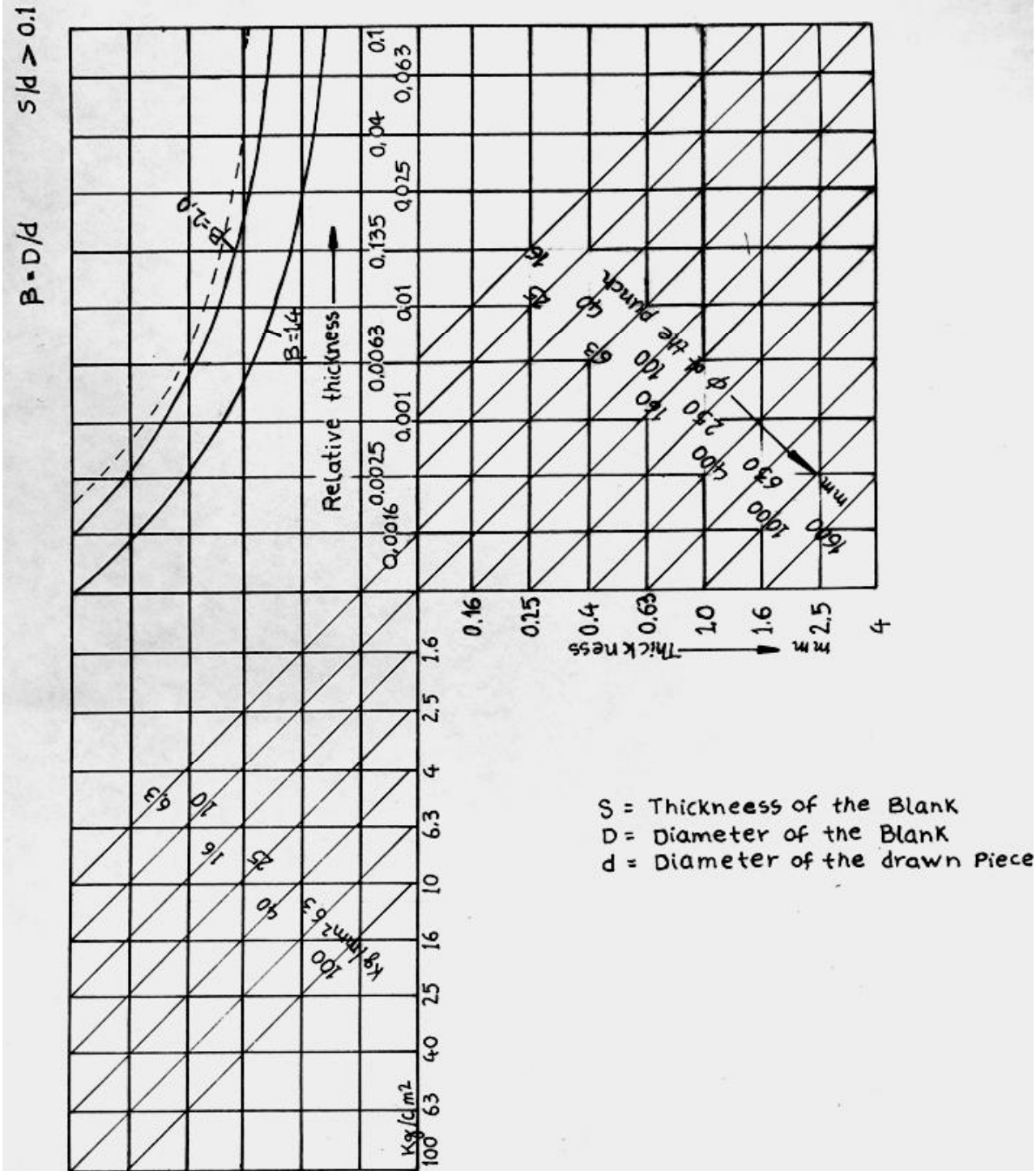
MATERIAL = ALUMINIUM

MATERIAL GRADE	'O' (ANNEALED)		H12 AND H32		H14 AND H34	
THICKNESS	0.5 - 3.2	> 3.2	0.5 - 3.2	> 3.2	0.5 - 3.2	> 3.2
% 1 st DRAW	40 - 42	42 - 45	32 - 34	34 - 36	25 - 28	22 - 28
% 2 nd DRAW	14 - 22	20 - 28	15	15	15	15
% 3 rd DRAW	14 - 22	14 - 22	15	15	15	15
% 4 th DRAW	14 - 22	14 - 22	15	15	15	15
MATERIAL	H16 ; H36 ; H18 ; H38		61S T4 AND R301-T3		6061 T6 AND 2024-T3	
THICKNESS	0.5 - 3.2	> 3.2	0.5 - 3.2	> 3.2	0.5 - 3.2	> 3.2
% 1 st DRAW	20 - 22	22 - 24	24 - 28	28 - 32	20 - 22	22 - 25
% 2 nd DRAW	NOT RECOMMENDED		NOT RECOMMENDED		NOT RECOMMENDED	
% 3 rd DRAW	NOT RECOMMENDED		"		"	
% 4 th DRAW	NOT RECOMMENDED		"		"	

DRAW RADIUS DIAGRAM

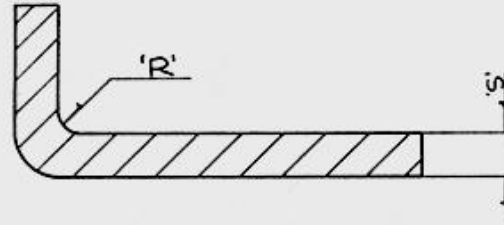


PRESSURE PAD'S PRESSURE



BEND

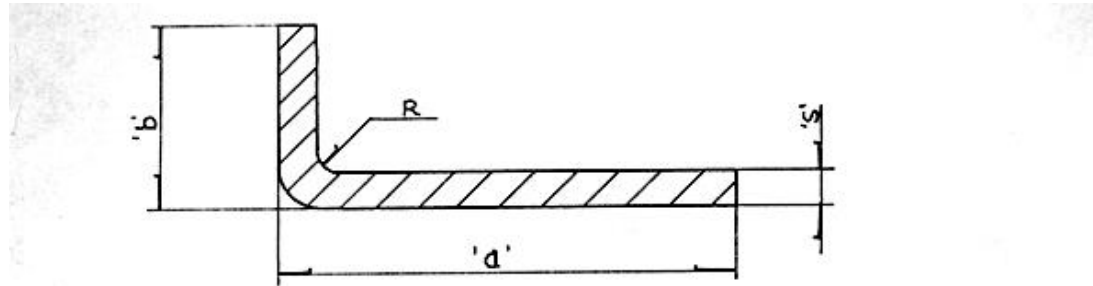
MINI ALLOWABLE BEND RADIUS FOR DIFERENT MATERIAL.



ALL DIMENSIONS ARE IN 'MM'

THICKNESS 'S'	0.1	0.3	0.5	0.8	1.0	1.5	2.0	3.0	4.0	6.0	8.0	10.0
MATERIAL	MINIMUM BENDING RADIUS = R											
COPPER (SOFT)	0.1	0.1	0.1	0.2	0.2	0.25	0.3	0.5	0.7	1.3	2.0	2.5
BRASS (SOFT)	0.1	0.15	0.2	0.25	0.3	0.4	0.5	0.7	1.0	2.0	3.5	5.5
BRASS (HARD)	0.1	0.2	0.3	0.4	0.5	0.6	0.8	1.2	1.5	2.5	4.0	6.0
STEEL (SOFT)	0.1	0.2	0.3	0.4	0.5	0.6	0.9	1.4	2.0	3.2	4.5	6.0
STEEL (MED HARD)	0.1	0.2	0.3	0.4	0.5	0.6	1.0	1.5	2.5	4.5	7.0	10.0
Alcu Mg	0.2	0.4	0.8	1.2	2.0	3.0	4.0	6.0	8.0	12.0	14.0	16.0
MgAl.	0.5	1.0	2.0	4.0	6.0	9.0	12.0	18.0	24.0	35.0	50.0	60.0

BEND ALLOWANCE FOR 90° BEND.



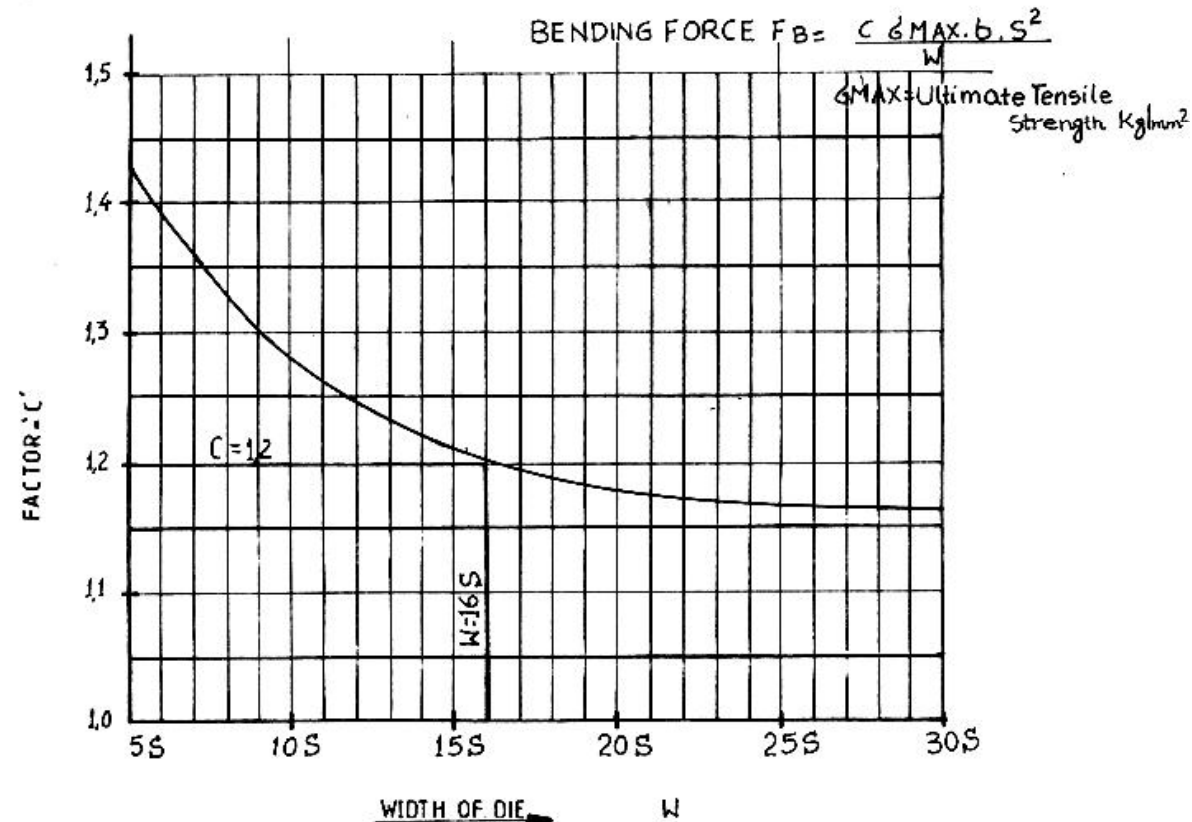
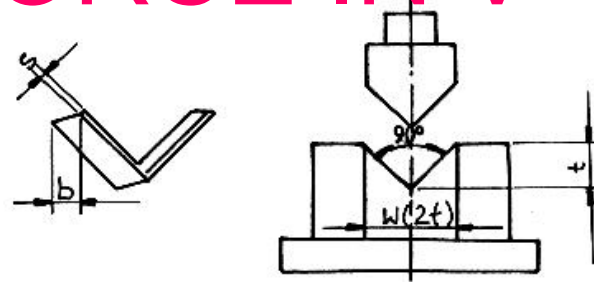
DEVELOPED LENGTH (90° BEND) $L = a + b - x$

FOR X DIMENSIONS REFER TABLE

ALL DIMENSIONS ARE 'mm'

THICKNESS-S	0.5	0.8	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	6.0	
X-DIMENSION													
RADIUS=R	0.1	0.7	1.2	1.4	2.1	2.8	3.5	4.3	5.1	5.8	6.5	7.4	8.6
	0.5	0.9	1.4	1.6	2.3	3.0	3.7	4.5	5.2	5.9	6.7	7.5	8.7
	1.0	1.2	1.6	1.8	2.5	3.2	3.9	4.7	5.4	6.1	6.9	7.7	8.9
	1.5	1.4	1.8	2.0	2.7	3.4	4.1	4.8	5.5	6.3	7.1	7.9	9.1
	2.0	1.7	2.0	2.3	2.9	3.6	4.3	5.0	5.7	6.5	7.3	8.1	9.3
	2.5	1.9	2.3	2.5	3.1	3.8	4.5	5.2	5.8	6.6	7.5	8.3	9.5
	3.0	2.1	2.5	2.7	3.3	4.0	4.7	5.3	6.0	6.8	7.7	8.5	9.8
	3.5	2.3	2.7	2.9	3.5	4.2	4.9	5.5	6.2	7.0	7.8	8.7	10.0
	4.0	2.6	3.0	3.2	3.7	4.4	5.0	5.7	6.3	7.2	8.0	8.9	10.3
	4.5	2.8	3.2	3.4	4.0	4.6	5.2	5.8	6.5	7.4	8.2	9.1	10.5
	5.0	3.0	3.4	3.6	4.2	4.8	5.4	6.0	6.7	7.5	8.4	9.3	10.7

BENDING FORCE IN V - DIE



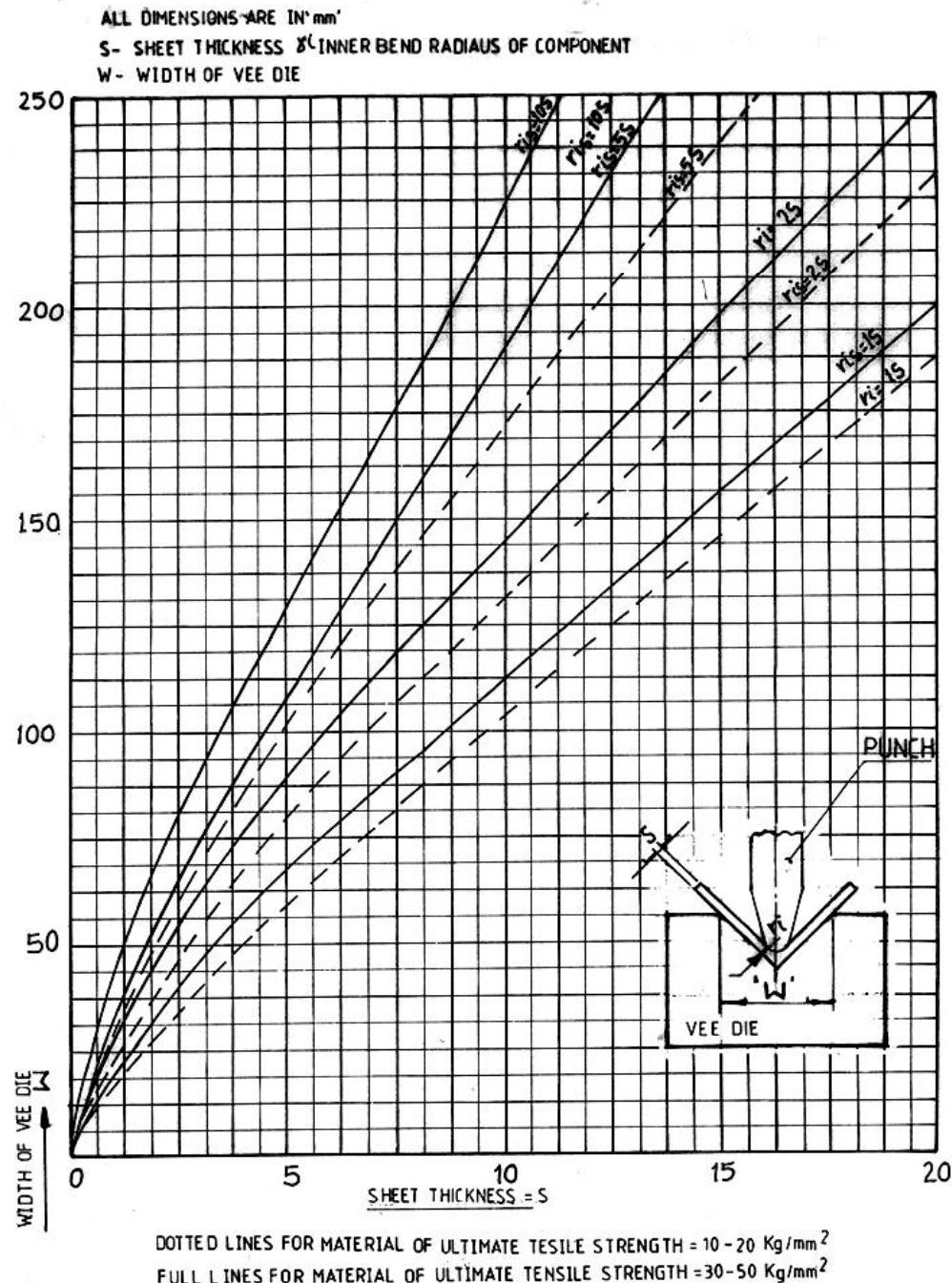
S = SHEET THICKNESS; C = FACTOR; W = WIDTH OF 'V' - DIE

b = WIDTH/BREATH OF THE SHEET. DIMENSIONS ARE IN 'mm'

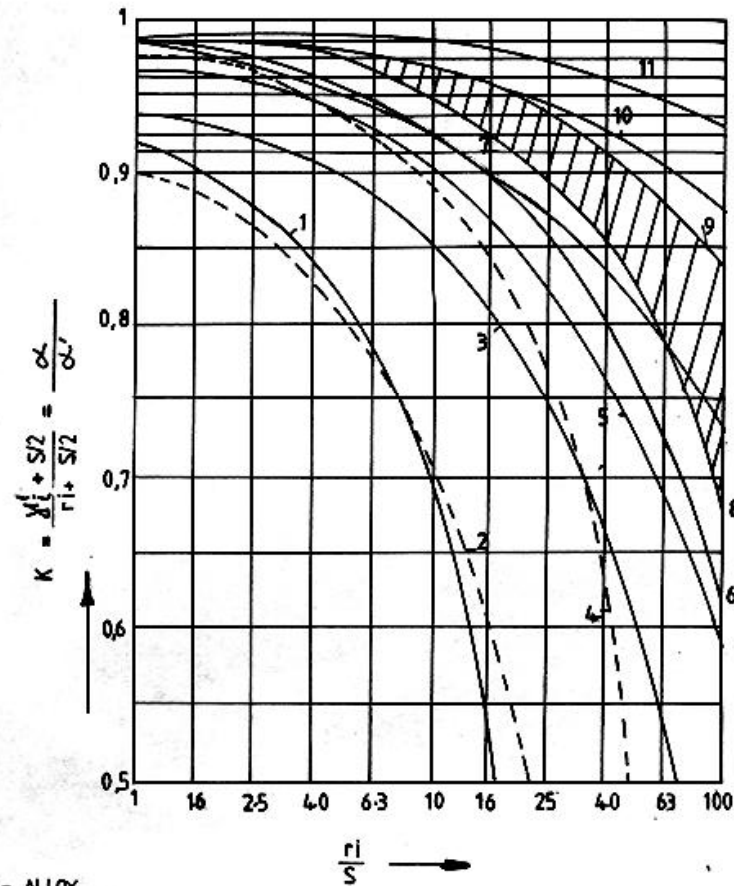
σ_{max} - ULTIMATE TENSILE STRENGTH Kg/mm^2 ; t = DEPT OF DIE

$$BENDING\ FORCE\ F_B = \frac{C \times \sigma_{max} \times b \times s^2}{W} \text{ Kg}$$

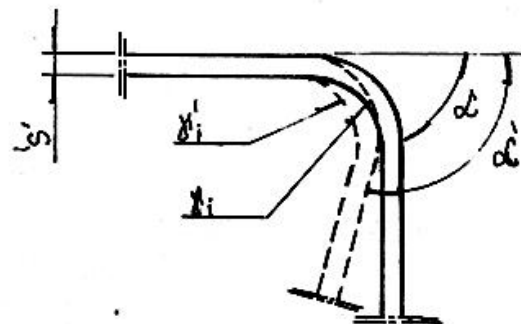
WIDTH OF A V – DIE FOR SHEET THICKNESS AND RADIUS.



SPRING BACK IN BENDING.



- | | |
|-------------------------|----------------------------------|
| 1 Al Cu Mg ALLOY | 7 Brass 67 soft |
| 2 St 18/8Cr Ni annealed | 8 AlZnCuMgSi Alloy (St I-V 23) |
| 3 Al Zn Cu Mg annealed | 9 MS CRCA deep drawing qualities |
| 4 St 18/8 Cr Ni hard | 10 AlCuMg & AlZnCuMg Soft alloy |
| 5 Al Mg Si | 11 Al Soft |
| 6 Cu 1/4 hard | |



- S = SHEET THICKNESS
 K = SPRING BACK FACTOR
 α = REQUIRED COMP. ANGLE
 Y_i = REQUIRED COMP RADIUS
 α' = BEND ANGLE BEFORE SPRING BACK
 r_i = INNER RADIUS

BEND ALLOWANCE FOR 90° BEND FOR STEEL

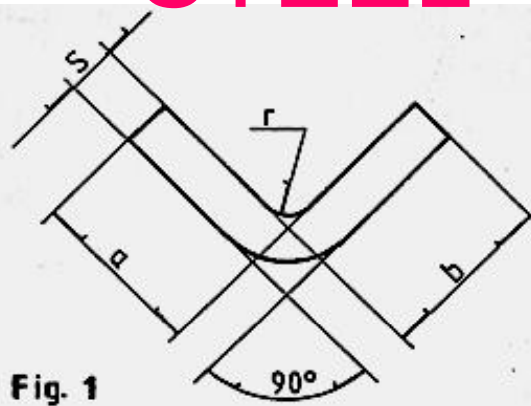


Fig. 1

All dimensions in 'mm'

BEND RADIUS 'r'	MINUS	PLUS												
	0.3	0.5	0.8	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0
0.25	0.03	0.10	0.27	0.38	0.65	0.95	1.26	1.55	1.87	2.16	2.48	2.68	2.92	3.09
0.50	0.13	0.00	0.17	0.28	0.56	0.86	1.18	1.49	1.83	2.14	2.44	2.66	2.90	3.08
0.75	0.23	0.11	0.06	0.18	0.46	0.76	1.09	1.42	1.77	2.09	2.41	2.62	2.87	3.06
1.00	0.33	0.22	0.04	0.07	0.36	0.68	1.00	1.35	1.72	2.04	2.36	2.59	2.84	3.04
1.25	0.44	0.32	0.14	0.03	0.26	0.58	0.92	1.27	1.65	1.99	2.32	2.55	2.81	3.02
1.50	0.54	0.43	0.25	0.13	0.16	0.50	0.84	1.20	1.58	1.93	2.26	2.51	2.77	2.99
1.75	0.64	0.53	0.35	0.23	0.07	0.40	0.75	1.12	1.51	1.87	2.20	2.46	2.72	2.96
2.00	0.74	0.64	0.46	0.32	0.04	0.32	0.66	1.04	1.44	1.80	2.14	2.41	2.68	2.92
2.50	0.96	0.85	0.67	0.54	0.24	0.12	0.47	0.87	1.28	1.65	2.00	2.30	2.58	2.84
3.00	1.18	1.06	0.87	0.75	0.44	0.07	0.27	0.70	1.10	1.49	1.85	2.15	2.48	2.75
3.50	1.41	1.28	1.09	0.97	0.65	0.29	0.09	0.51	0.92	1.31	1.68	2.01	2.33	2.64
4.00	1.66	1.54	1.35	1.20	0.87	0.51	0.10	0.30	0.71	1.12	1.50	1.84	2.16	2.50
4.50	1.93	1.78	1.58	1.44	1.11	0.75	0.34	0.08	0.48	0.92	1.30	1.84	1.98	2.32
5.00	2.22	2.06	1.88	1.73	1.40	1.03	0.62	0.22	0.22	0.66	1.04	1.40	1.74	2.08

BEND ALLOWANCE FOR 90° BEND FOR STEEL

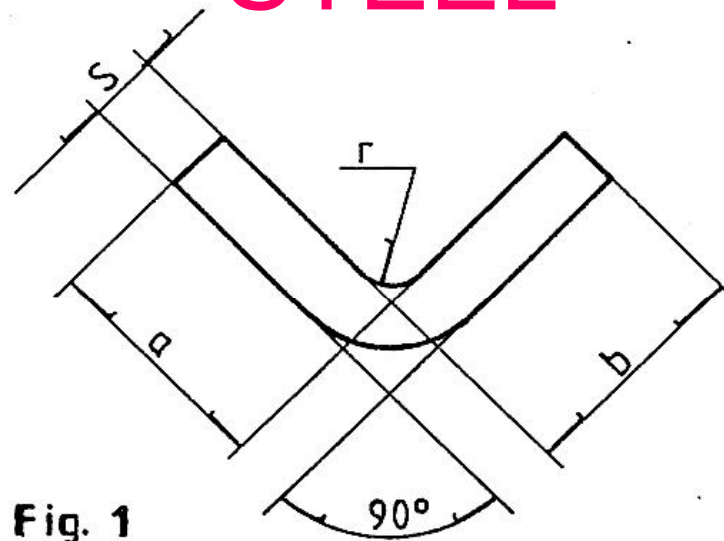


Fig. 1

DESCRIPTION : This table is applied to 90° bend on steel material. For inner radius of 'r', the neutral fibre length may be lengthened or shortened, depending on the thickness of the sheet

TOTAL DEVELOPED LENGTH: Will be the sum of the inner (a + b) dimensions of the bend and plus or minus of the neutral fibre length as per the table. REF FIG 1

NOTE : All dimensions above the thicker lines, are to be added with 'a + b', to get the developed length. REF. FIG 1

All dimensions below the thicker lines, are to be subtracted from 'a + b', to get the developed length. REF FIG 1

BEND ALLOWANCE FOR 90° BEND FOR STEEL.

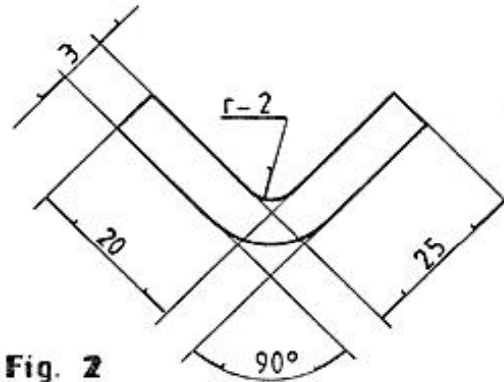


Fig. 2

REF FIG 2 AND CHART (SH.1)

EXAMPLE 1 For sheet thickness of 3mm and inner bend radius of 2mm;

According to the chart = + 1.04

ie. length to add = 1.04

$$\begin{aligned}\therefore \text{Total length} &= 20 + 25 + 1.04 \\ &= \underline{46.04}\end{aligned}$$

REF FIG 2 AND CHART (SH.1)

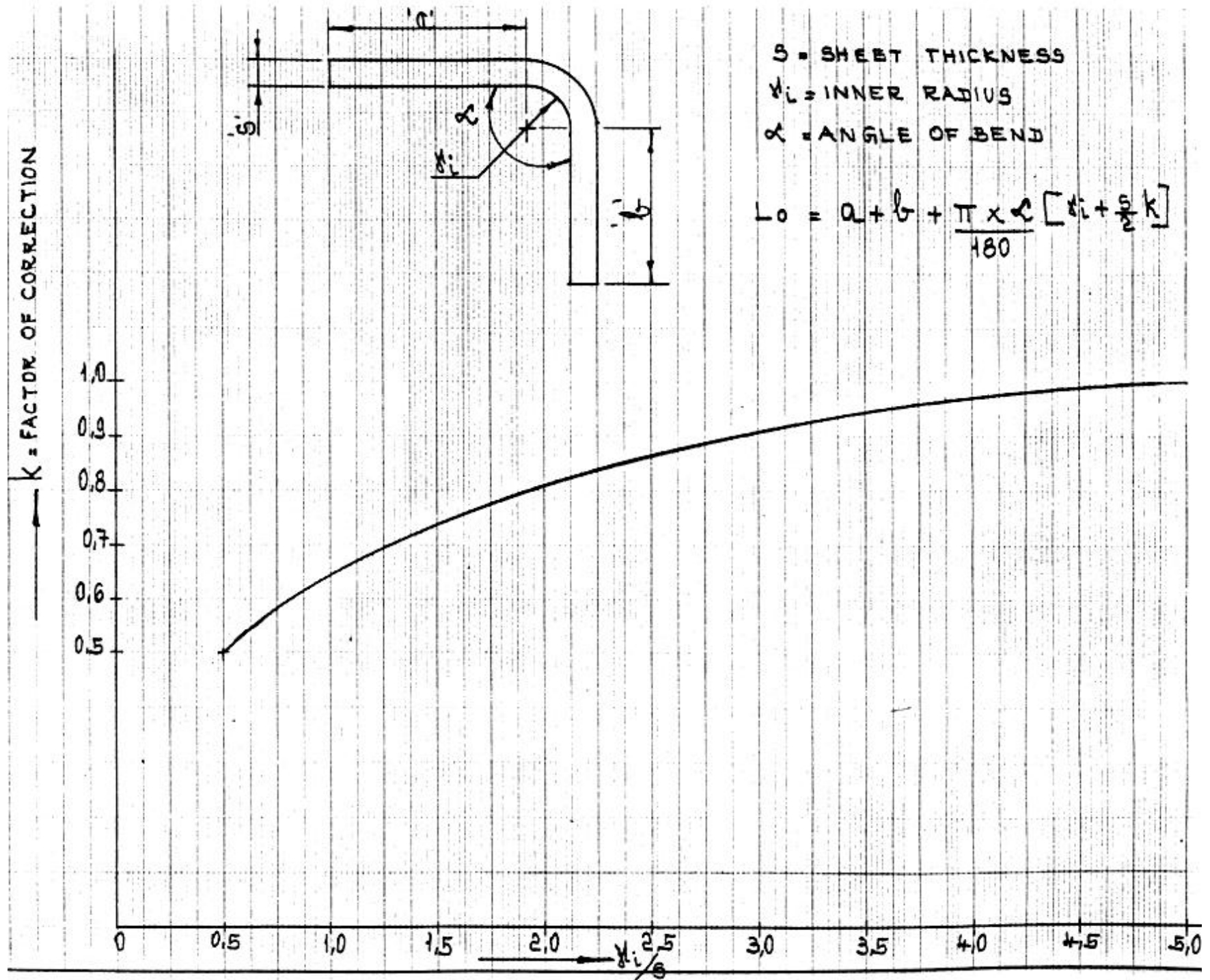
EXAMPLE 2 For sheet thickness of 1.5mm and inner bend radius of 2mm;

According to chart = - 0.04 mm

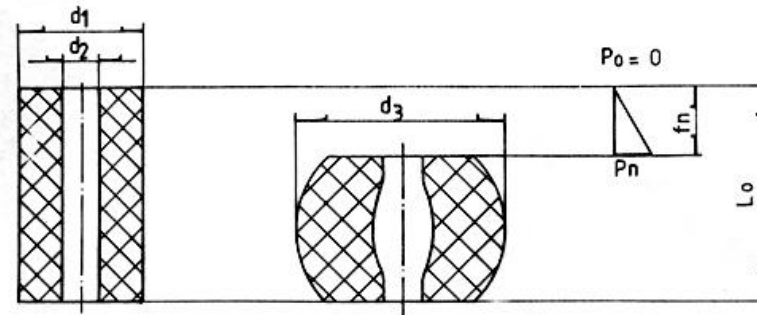
ie. length to subtract = 0.04

$$\begin{aligned}\therefore \text{Total length} &= 20 + 25 - 0.04 \\ &= \underline{44.96}\end{aligned}$$

NEUTRAL FIBRE FACTOR



PUR SPRING

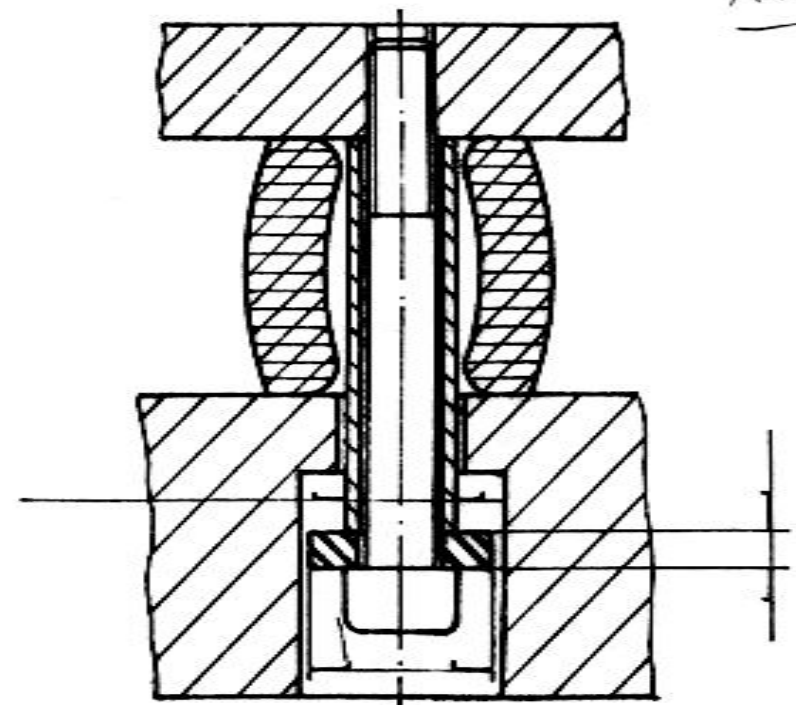
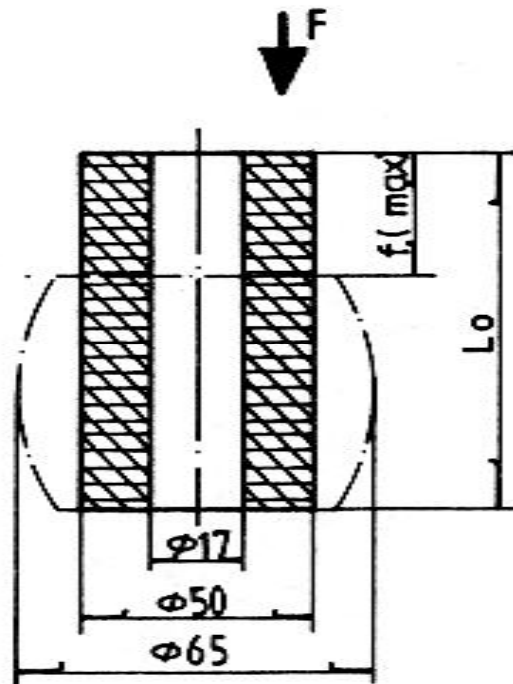


NEOPRENE

HARDNESS = 68 ± 5 Shore

			$P = KP$			f_n (mm)			d_3 (mm)					
d_1 mm	d_2 mm	L_0	$f_n = 0.1 L_0$			$f_n = 0.2 L_0$			$f_n = 0.3 L_0$			$f_n = 0.35 L_0$		
			P	F_n	d_3	P	f_n	d_3	P	f_n	d_3	P	F_n	d_3
16	6.5	20	9	2	18	15	4	19	25	6	21	30	7	22
20	8.5	25	13	2.5	22	25	5	24	45	7.5	26	50	8.75	27
		32		3.2			6.4			9.6			11.2	
25	8.5	25	22	2.5	28	45	5	30	75	7.5	33	80	8.75	34
		32		3.2			6.4			9.6			11.2	
		40		4			8			12			14	
32	13.5	32	45	3.2	35	90	6.4	38	150	9.6	42	180	11.2	43
		40		4			8			12			14	
		50		5			10			15			17.5	
		63		6.3			12.6			18.9			22	
40	13.5	32	80	3.2	43	160	6.4	47	270	9.6	51	380	11.2	53
		40		4			8			12			14	
		50		5			10			15			17.5	
		63		6.3			12.6			18.9			22	
		80		8			16			24			28	
50	17	32	120	3.2	53	270	6.4	58	440	9.6	63	630	11.2	65
		40		4			8			12			14	
		50		5			10			15			17.5	
		63		6.3			12.6			18.9			22	
		80		8			16			24			28	
		100		10			20			30			35	
63	17	32	190	3.2	67	420	6.4	73	700	9.6	79	930	11.2	82
		40		4			8			12			14	
		50		5			10			15			17.5	
		63		6.3			12.6			18.9			22	
		80		8			16			24			28	
		100		10			20			30			35	
80	21	50	370	5	87	780	10	95	1300	15	102	1420	17.5	106
		63		6.3			12.6			18.9			22	
		80		8			16			24			28	
		100		10			20			30			35	
100	21	40	350	4	110	1200	8	120	1800	12	130	2900	14	135
		63		6.3			12.6			18.9			22	
		100		10			20			30			35	

PUR SPRING



Spring dia. 50 mm 80 Shore

Lo in mm	fmax in mm	Fmax in N
32	10.6	10800
40	14	10400
50	17.5	10200
63	22	10800

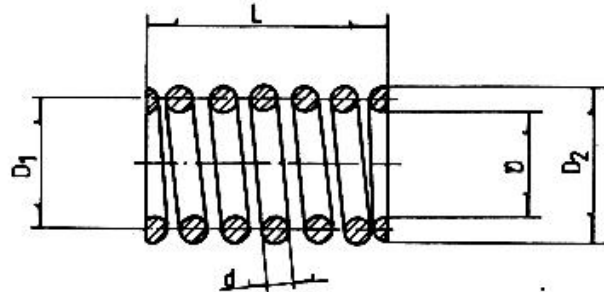
Spring dia. 50mm 90 Shore

Lo in mm	fmax in mm	Fmax N
32	9.6	17400
40	12	17300
50	15	17000
63	18.9	16650

Spring dia. 50 mm 95 Shore

Lo in mm	fmax in mm	Fmax in N
32	8	13000
40	10	20100
50	12.5	19600
63	15.7	19200

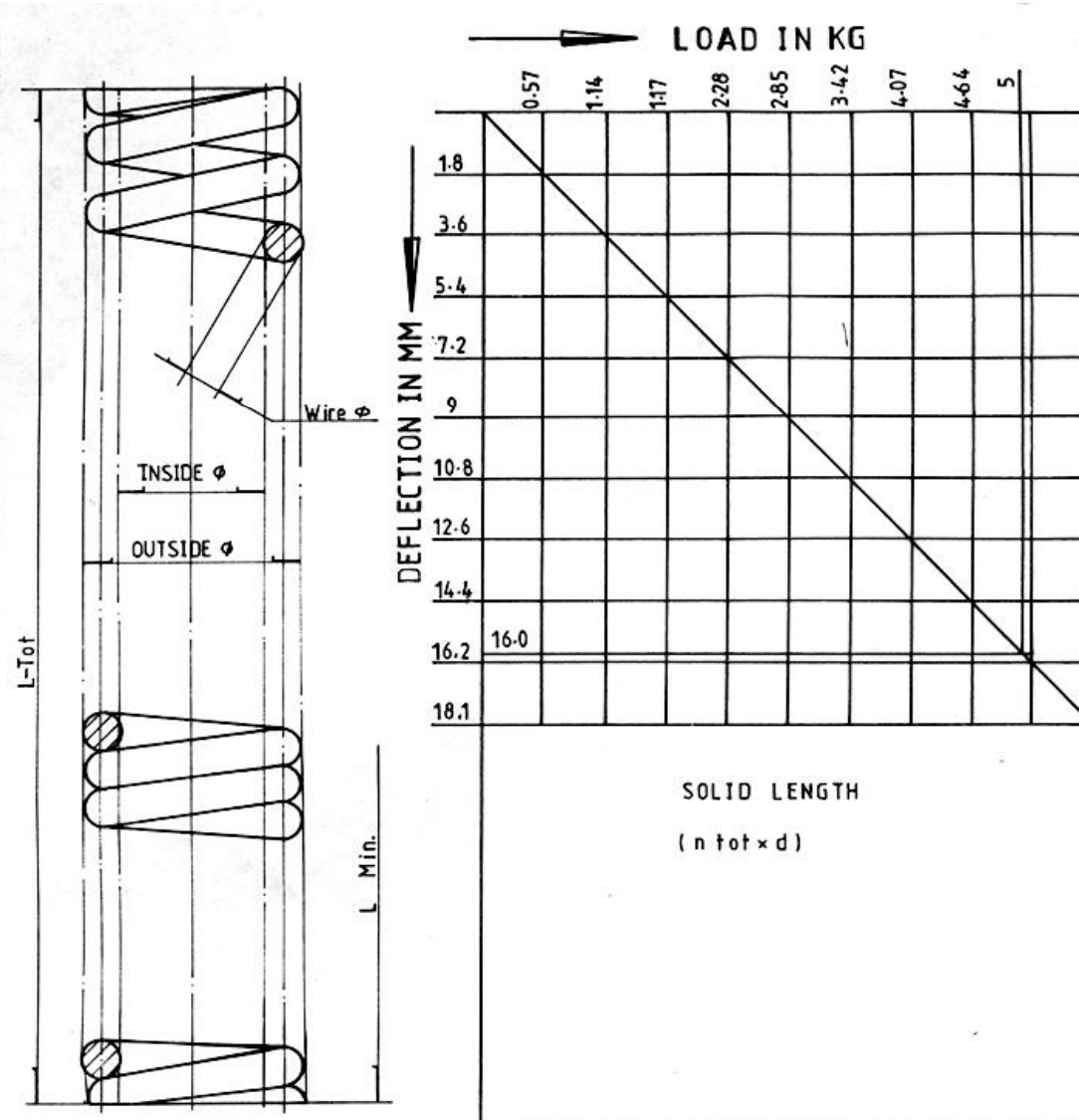
COMPRESSION SPRING



NO	d	D	D ₁	D ₂	Free L	Stroke max	(Kg) Load max	Total No of coils	
1	1.5	7	8.5	10	10	2.7	11	4.25	
2	1.5	9	10.5	12	25	12.5	9	7.5	
3	1.5	11	12.5	14	50	32	9	11.3	
4	2	7	9	11	15	4	22	5	
5	2	9	11	13	35	14	19	10	
6	2	11	13	15	50	25	17	12	
7	2.5	9	11.5	14	25	7	32	6.8	
8	2.5	15	17.5	20	50	25	23	8.5	
9	3	9	12	15	15	2.6	48	4	
10	3	11	14	17	35	9.7	43	8.1	
11	3	13	16	19	50	17.5	40	10.5	
12	3.5	11	14.5	18	25	5.1	62	5.5	
13	3.5	15	18.5	22	50	17	52	9	
14	4	9	13	17	25	3	90	5.3	
15	4	11	15	19	50	9	80	10	
16	4	21	25	29	70	29	57	10	
17	4.5	15	19.5	24	35	7.4	94	6	
18	4.5	26	30.5	35	50	21.5	65	6	
19	5	11	16	21	25	2.6	135	4.3	
20	5	15	20	25	50	10	118	7.8	
21	5	21	26	31	80	25	100	10.5	
22	6	13	19	25	35	3.5	180	5	
23	6	26	32	38	70	20	130	8	
24	7	21	28	35	50	7.8	210	5.8	
25	7	26	33	40	35	6.6	182	3.9	
26	7	31	38	45	70	20	168	7	
27	8	26	34	42	50	8.7	260	5	
28	8	31	39	47	100	25	230	9	

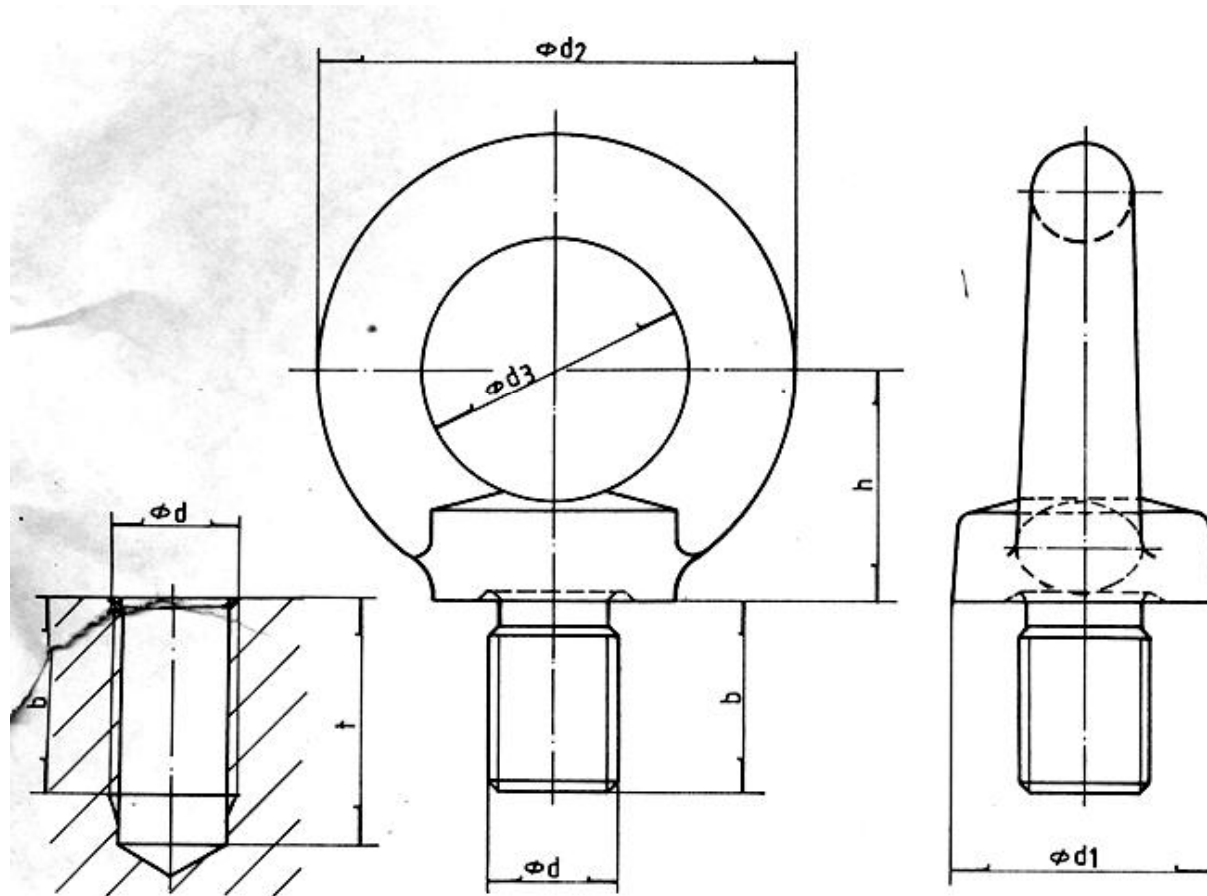
ALL DIMENSIONS ARE IN mm

COMPRESSION SPRING



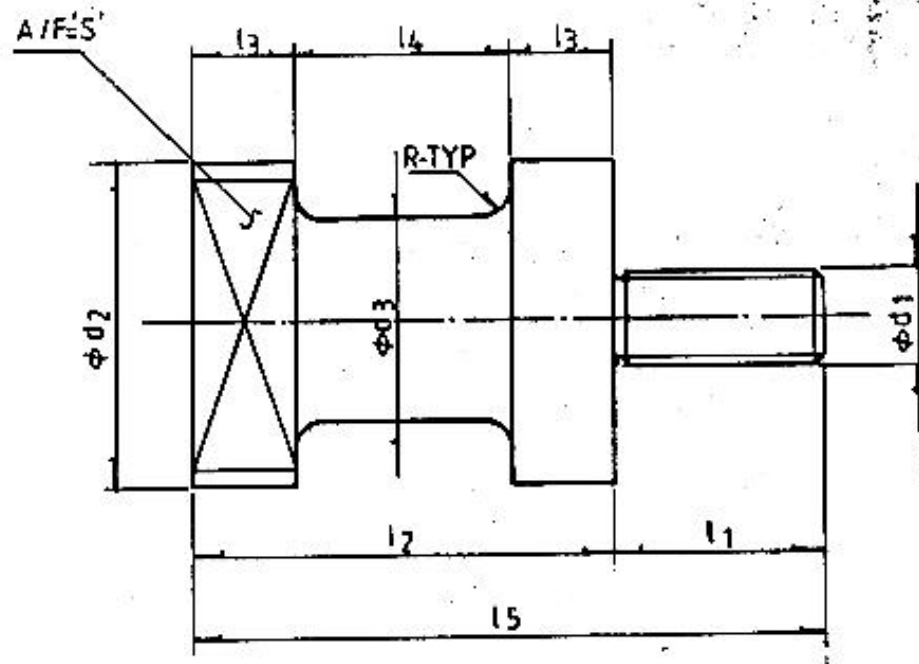
Wire ϕ = 1.2 mm
 Inside ϕ = 9.4 mm
 Outside ϕ = 11.8 mm
 Total No of coils = 7 Eff No of coils = 5.5
 Total length 26.5
 Maximum permissible stroke = 16 mm
 Maximum permissible load = 5 kg
 Torque stress at F max = 95 Kg/mm²
 Material Spring steel

EYE – BOLT



ϕd	M 10	M 12	M 16	M 20	M 24	M 30	M 36	M 42
b	18	22	28	30	38	45	55	65
ϕd_1	25	30	35	40	50	65	75	85
ϕd_2	45	54	63	72	90	108	126	144
ϕd_3	25	30	35	40	50	60	70	80
h	22	26	30	35	45	55	65	75
t	24	29	35	38	47	55	66	77

PLATE LIFTING BOLT

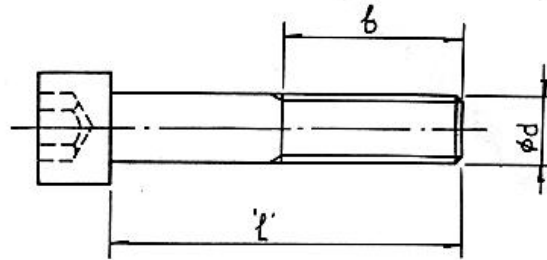


GEN TOL = $\pm 0,1$

TYPE	ϕd_1	ϕd_2	ϕd_3	l_1	l_2	l_3	l_4	l_5	R	A/F 'S'	
1	M8	25	15	15	50	12	26	65	2	21	
2	M10	30	20	20	50	12	26	70	2	27	
3	M12	40	25	25	50	12	26	75	3	36	
4	M16	50	30	30	50	12	26	80	3	46	
5	M20	60	35	35	50	12	26	85	4	55	

TO ORDER A LIFTING BOLT OF $\phi d = M12$ WRITE : 00181 20/TYPE - 3

THREAD LENGTHS FOR ALLEN SCREWS



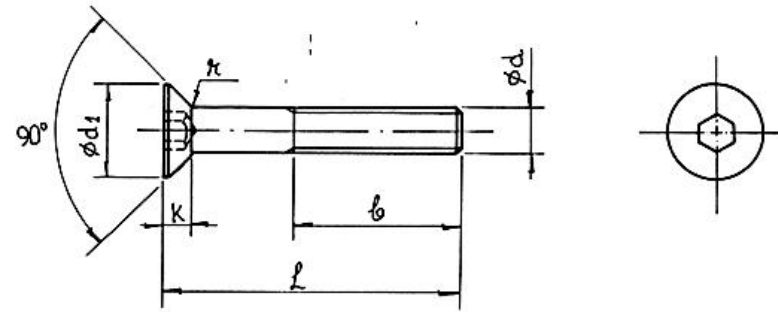
ALL DIMENSIONS ARE 'mm'

THREAD DIA ϕd	THREAD LENGTH = b		
	$l \leq 125$	$125 < l \leq 200$	$l > 200$
1,6	9	—	—
2	10	—	—
2,5	11	—	—
3	12	—	—
4	14	—	—
5	16	—	—
6	18	—	—
7	20	—	—
8	22	28	—
10	26	32	—
12	30	36	—
14	34	40	—
16	38	44	57
18	42	48	61
20	46	52	65
22	50	56	69
24	54	60	73
27	60	66	79
30	66	72	85
33	72	78	91
36	78	84	97

STANDARD LENGTHS FOR SCREWS, BOLTS AND STUDS.		
2	35	140
2,5	(38)	150
3	40	160
4	45	170
5	50	180
6	55	190
(7)	60	200
8	65	220
(9)	70	240
10	75	260
(11)	80	280
12	85	300
14	90	
16	(95)	
(18)	100	
20	(105)	
(22)	110	
25	(115)	
(28)	120	
30	(125)	
(32)	130	

LENGTHS WITHIN BRACKETS SHOULD BE AVOIDED.
(Reference : ISO 888)

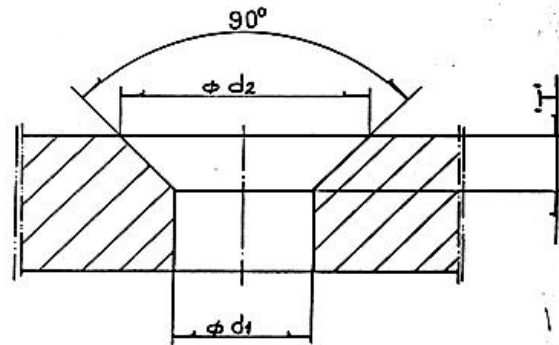
C / SK SCREWS STDS



SCREW SIZE ϕd		M3	M4	M5	M6	M8	M10	M12	(M14)	M16	(M18)	M20
b	1	12	14	16	18	22	26	30	34	38	42	46
	2				24	28	32	36	40	44	48	52
ϕd_1 (h14)		6	8	10	12	16	20	24	27	30	33	36
k		1.7	2.3	2.8	3.3	4.4	5.5	6.5	7.0	7.5	8.0	8.5
r_{min}		0.1	0.2	0.2	0.3	0.5	0.5	1	1	1	1	1
L'		LENGTH OF SCREWS										
8												
10												
12												
16												
18												
20												
25												
30												
35												
40												
45												
50												
55												
60												
70												
80												

PREFERRED SCREW SIZE AND LENGTHS ARE SHOWN WITHIN BOLD LINES.
 1- FOR LENGTHS BELOW 130 mm. 2- FOR LENGTHS ABOVE 130 mm.
 SIZES SHOWN WITH IN BRACKETS - NOT PREFERRED.

C / SK HOLES

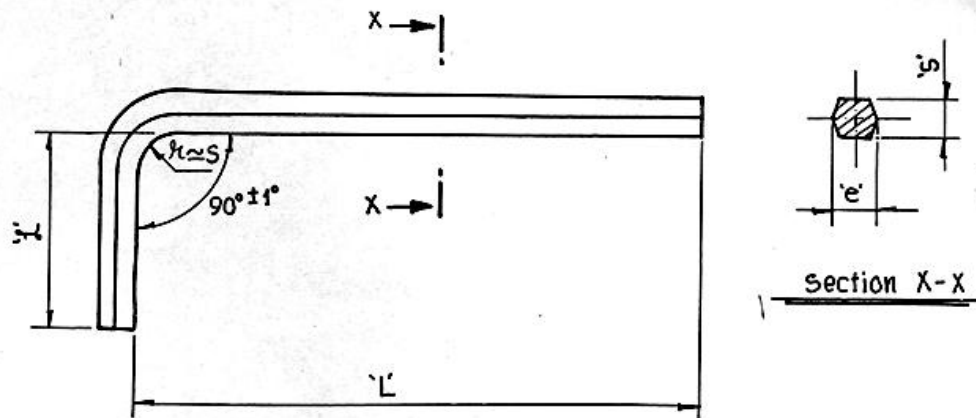


SCREW SIZE	MM COUNTERSUNK DETAILS		
	ϕd_1	ϕd_2	T
M2	2.4	4.5	1.2
M3	3.4	6.5	1.6
M4	4.5	8.6	2.1
M5	5.5	10.4	2.5
M6	6.6	12.4	2.9
M8	9	16.4	3.7
M10	11	20.4	4.7
M12	14	24.4	5.2
M14	16	27.4	5.7
M16	18	32.4	7.2
M18	20	36.4	8.2
M20	22	40.4	9.2

TO DESIGNATE - A COUNTER SUNK HOLE FOR M6 COUNTER
SUNK SCREW IS SPECIFIED AS

C/SK- M6

ALLEN KEY STDS



ALL DIMENSIONS 'mm'

NOMINAL SIZE = S	TOLERANCE ON 'S'	e		'L'	'ℓ'
		MAX	MIN		
1.5	$h9$	1.68	1.63	45	14
2	$h10$	2.25	2.18	50	16
2.5		2.82	2.75	56	18
3		3.39	3.31	63	20
4		4.53	4.44	70	25
5		5.67	5.58	80	28
6		6.81	6.71	90	32
8		9.09	8.97	100	36
10		11.37	11.23	112	40
12	$h11$	13.65	13.44	125	45
14		15.93	15.70	140	56
17		19.35	19.09	160	63
19		21.63	21.32	180	70
22		25.05	24.71	200	80
24		27.33	26.97	224	90
27		30.75	30.36	250	100
32		36.45	35.98	315	125
36		41.01	40.50	355	140

PREFERRED SIZES OF SCREWS, DOWELS, PIPE THREADS

PREFERED SIZES OF DOWELS

DIA SIZES mm	2	3	4	5	6	8	10	12	16	20
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PREFERED SIZES OF HEX. SOCKET HEAD SCRES(ALLEN SCREWS)

THREAD SIZES METRIC THD.	M3	M4	M5	M6	M8	M10	M12	M16	M20	M24
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PREFERED SIZES OF COUNTER SUNK SCREWS

THREAD SIZES METRIC THD.	M2	M3	M4	M5	M6	M8	M10	M12	M16	M20
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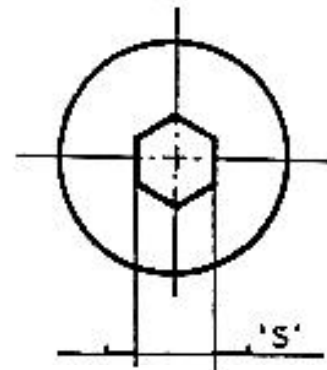
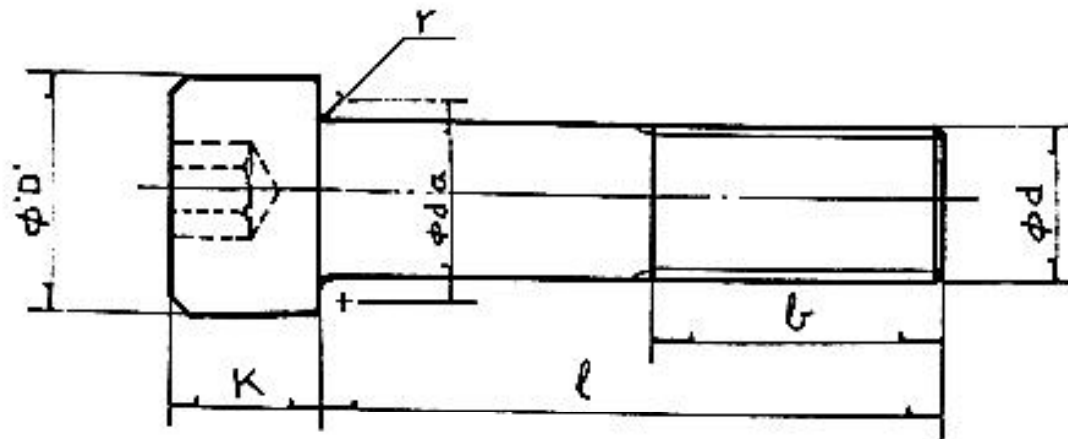
PREFERED SIZES OF SOCKET HEAD GRUB SCREWS

THREAD SIZES METRIC THD.	M6	M8	M10	M12	M16	M20	M24
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PREFERED SIZES OF PIPE THREADS

THREAD SIZES B.S.P - THD	1/8	1/4	3/8	1/2	3/4	1"	1 1/4	1 1/2	2"	2 1/4	2 1/2	3"
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ALLEN SCREWS STDS



SCREW SIZE ϕd	M3	M4	M5	M6	M8	M10	M12	(M14)	M16	(M18)	M20	(M22)	M24	(M27)	M30	(M33)	M36
l	12	14	16	18	22	26	30	34	38	42	46	50	54	60	66	72	78
					28	32	36	40	44	48	52	56	60	66	72	78	84
ϕD MAX	5.5	7	8.5	10	13	16	18	21	24	27	30	33	36	40	45	50	54
K	3	4	5	6	8	10	12	14	16	18	20	22	24	27	30	33	36
ϕda MAX	3.6	4.7	5.7	6.8	9.2	11.2	14.2	16.2	18.2	20.2	22.4	24.4	26.4	30.4	33.4	36.4	39.4
r	0.1	0.2	0.2	0.25	0.4	0.4	0.6	0.6	0.6	0.6	0.8	0.8	0.8	1	1	1	1
S	2.5	3	4	5	6	8	10	12	14	14	17	17	19	19	22	24	27

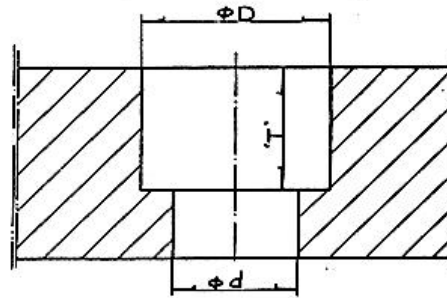
NOTE:- NON-PREFERRED SIZES SHOWN IN WITHIN BRACKETS

ALLEN SCREWS STDS

1. FOR LENGTHS UP TO 130 mm.
2. FOR LENGTHS ABOVE 130mm.
PREFERRED SCREW SIZES AND LENGTHS ARE SHOWN WITHIN BOLD LINES

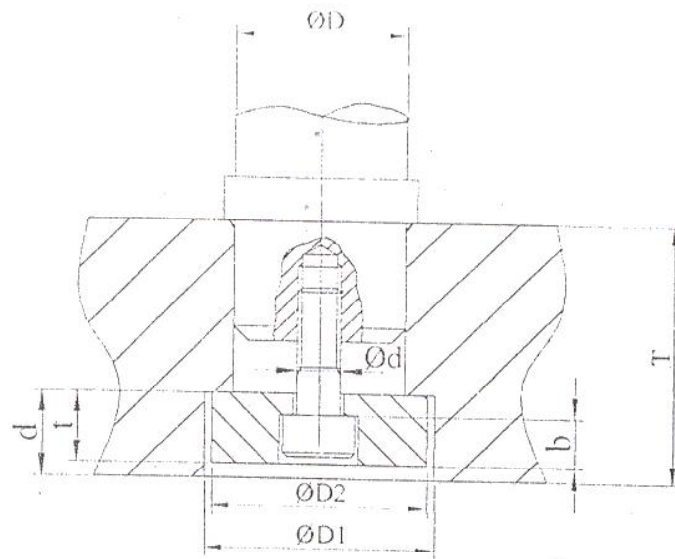
[illegible]

C / BORE FOR ALLEN SCREWS



ALLEN SCREW SIZE	COUNTER BORE DETAIL MM		
	ϕd	ϕD	T
M3	3.4	6	3.4
M4	4.5	8	4.6
M5	5.5	10	5.7
M6	6.6	11	6.8
M8	9.0	15	9
M10	11	18	11
M12	14	28	13
M14	16	24	15
M16	18	26	17.5
M18	20	30	19.5
M20	22	33	21.5
M22	24	36	23.5
M24	26	40	25.5
M27	30	43	28.5
M30	33	48	32

TO DESIGNATE : A COUNTER BORE HOLE, FOR M10 ALLEN SCREW
IS DESIGNATED AS
= C/BORE-M10



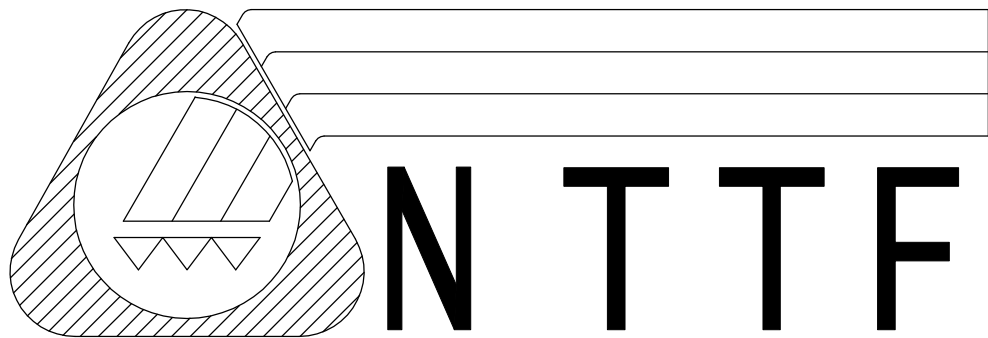
ØD	24/25	30/32	40/42
t	10	12	12
ØD2	30	36	46
ØD1	32	38	48
d	12	14	14
Ød	M6	M8	M8
b	6.5	8.5	8.5
T	36 / 46 / 56 / 66		

PART : PILLAR HOLDER.

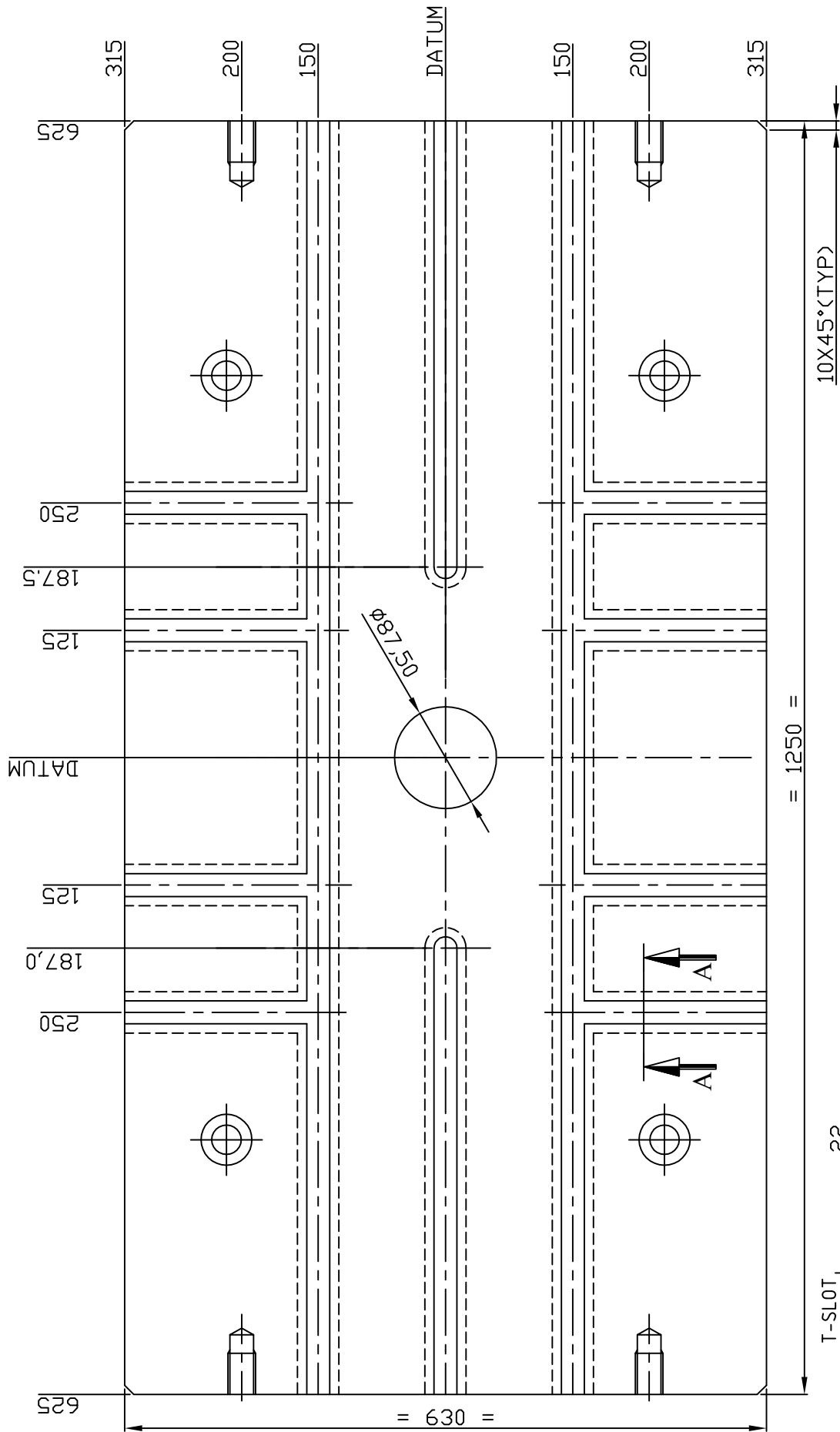
MATERIAL : MS

PRESS

BOLSTERS

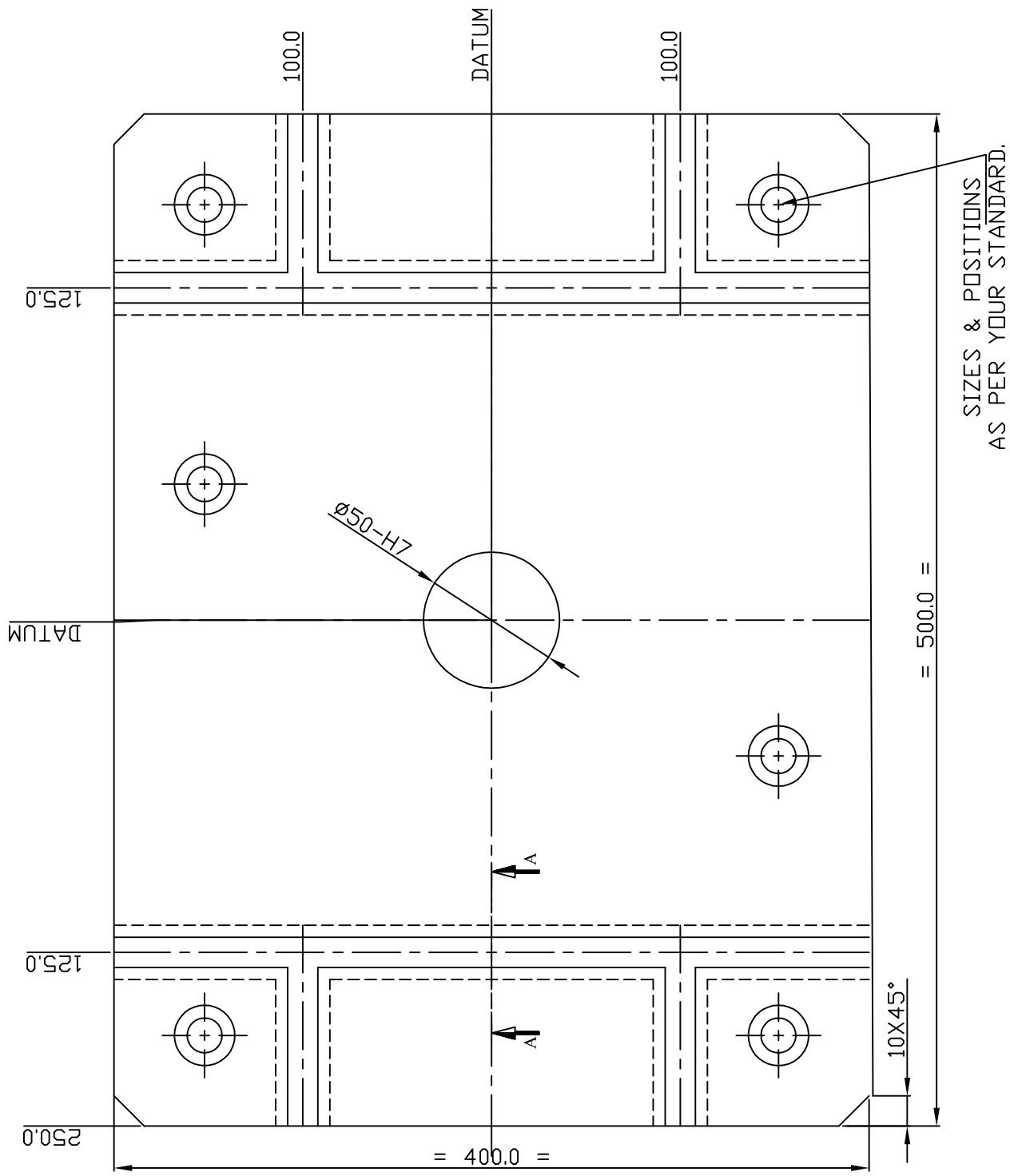


**COMPILED BY SRIVINAY
PGDTD 2009,
NTTF BANGALORE**

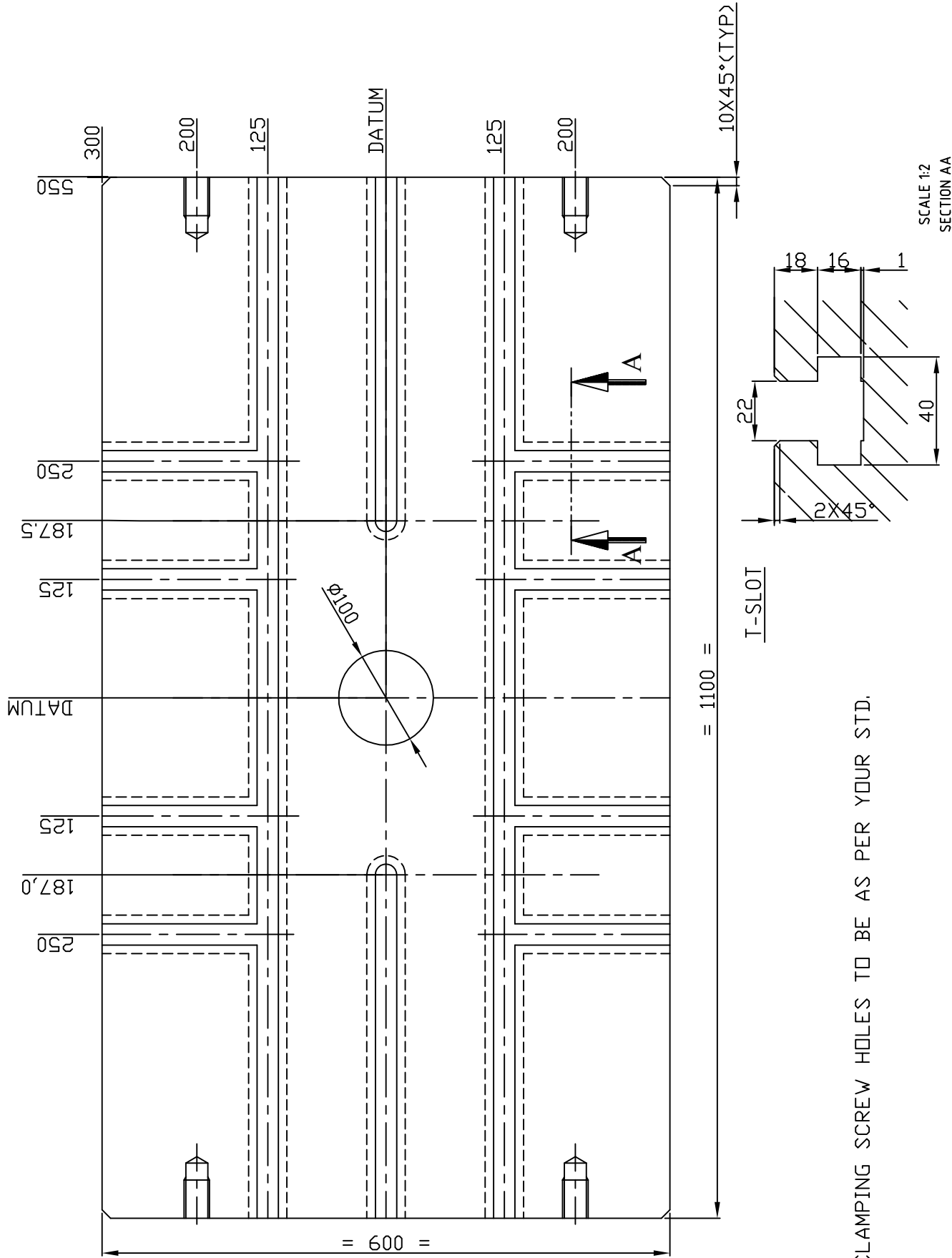


BOLSTER PLATE CLAMPING SCREW HOLES TO BE AS PER YOUR STD.

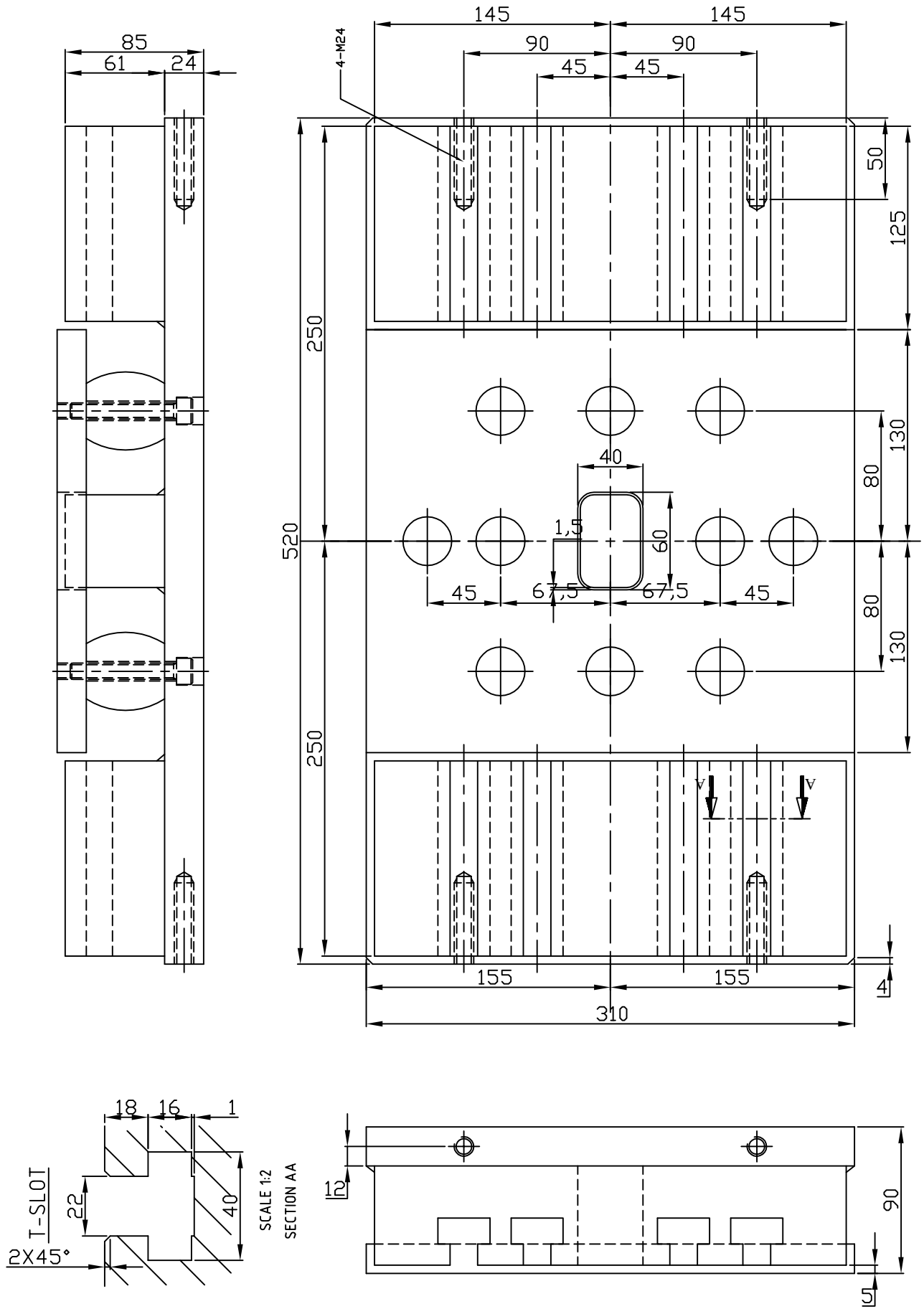
DRN	03 07 24	ANISH K		
CHD				
APPR				
PDTD'03-04 NTTF-TTC			160 T PRESS - II BOTTOM FACE	??



DRN	03 07 24	ANISH K		
CHD				
APPR				
PDTD'03-04 NTTF-TTC			63 T PRESS IV & V	??



DRN	03 07 24	ANISH K		
CHD				
APPR				
PDTD'03-04 NTTF-TTC			100 T PRESS -III BOTTOM FACE	??

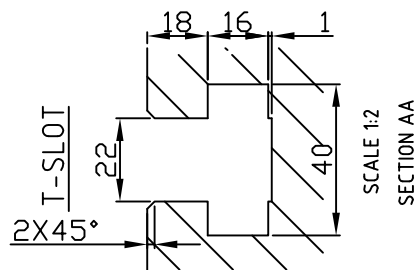
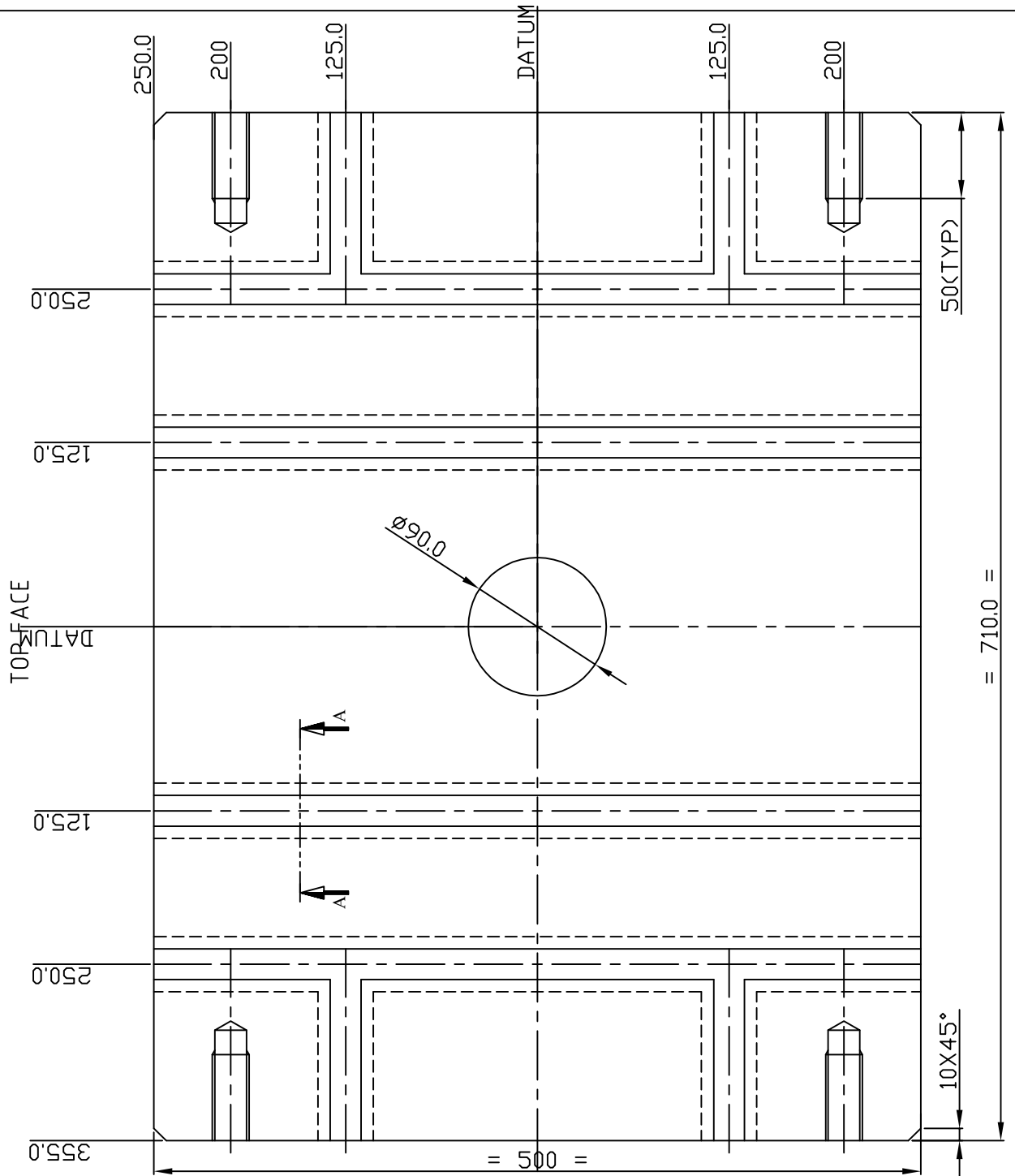


DRN	03 07 24	ANISH K
CHD		
APPR		

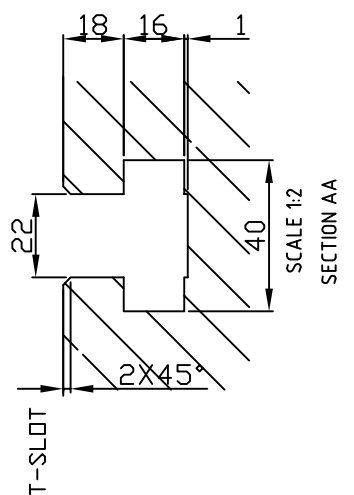
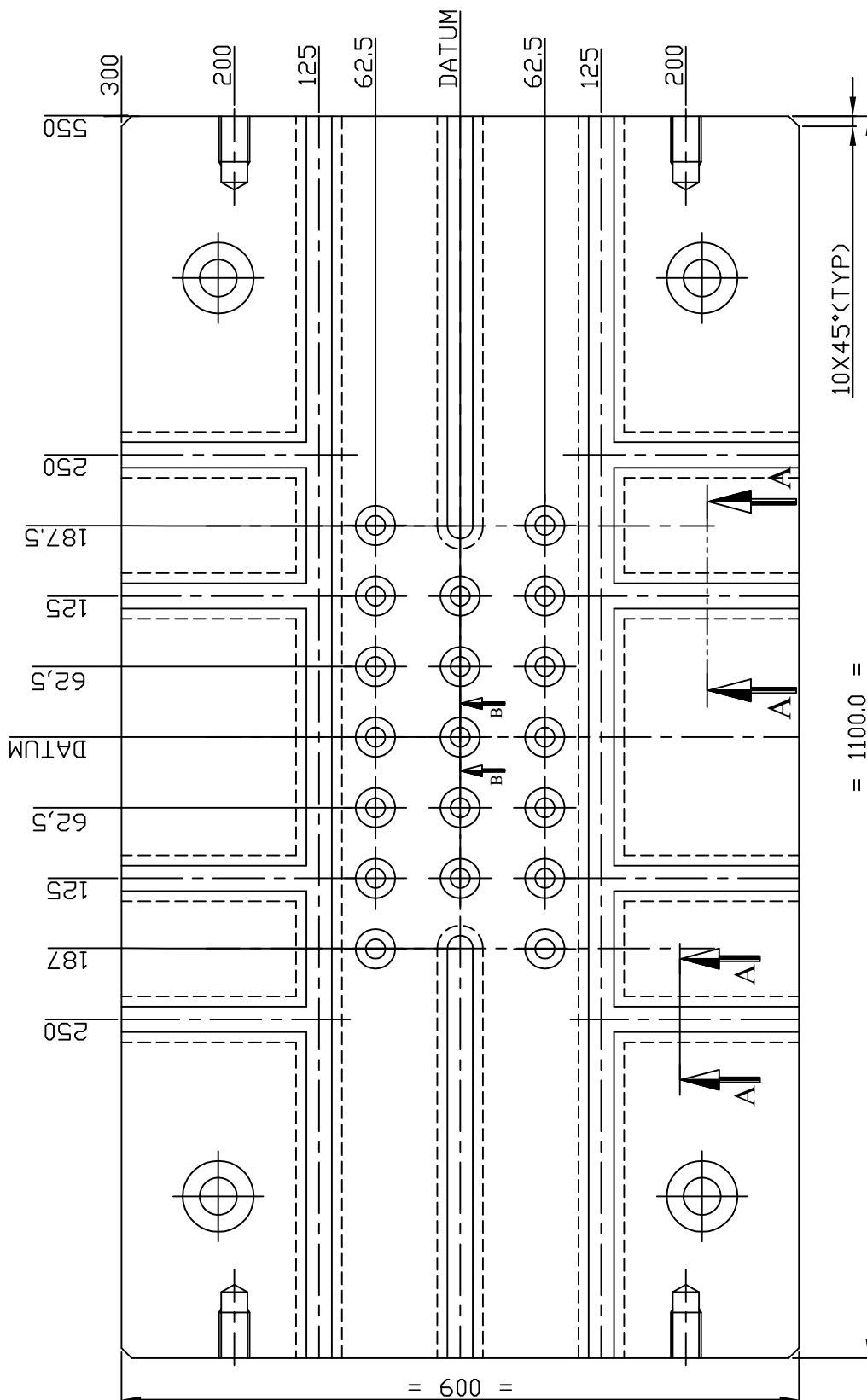
PDTD'03-04
NTTF-TTC

GODREJ 63 TON III (GEARED)

SLOTS FOR 160T PRESS SLIDE FACE - II

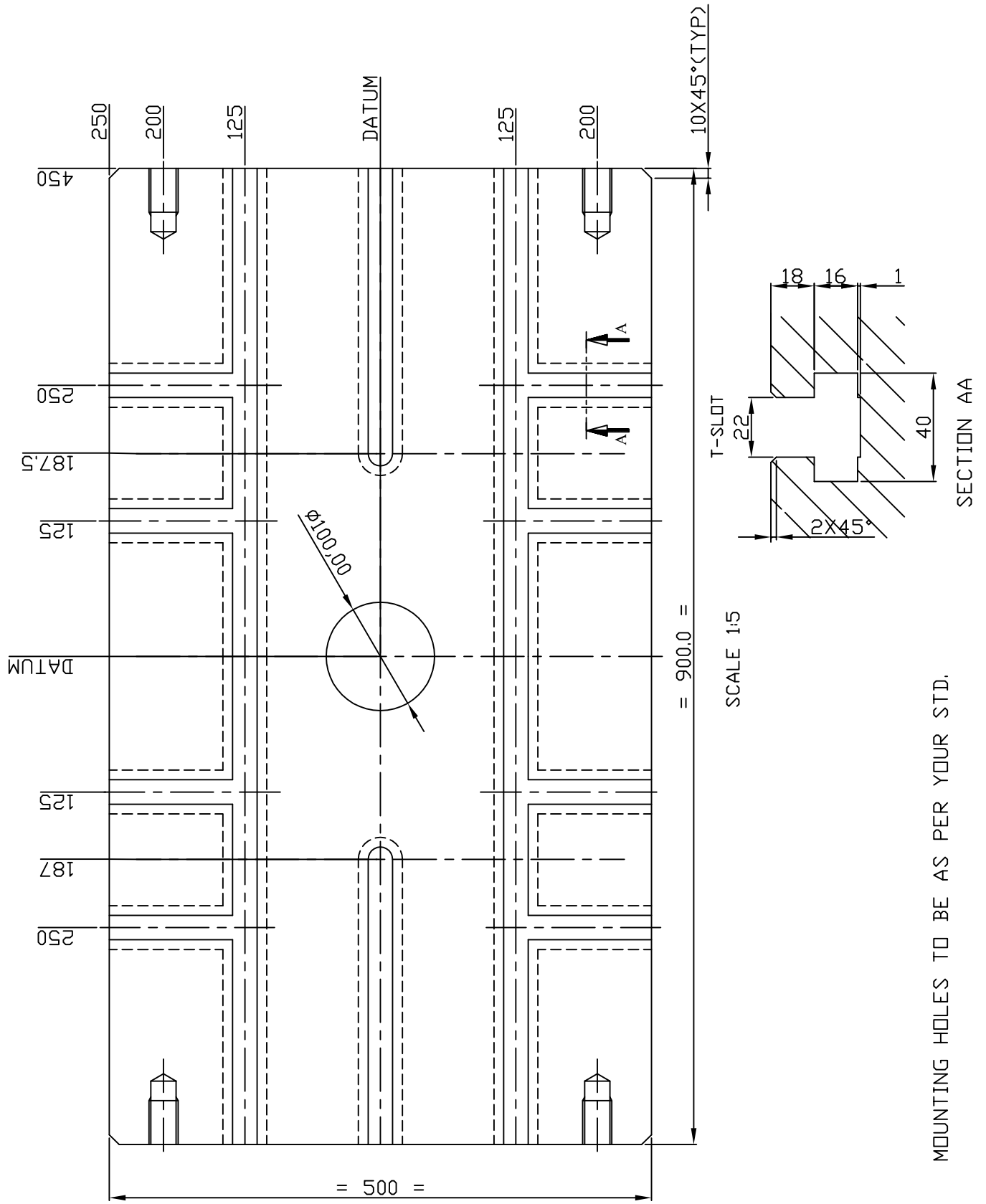


DRN	03 07 24	ANISH K		
CHD				
APPR				
PDTD'03-04 NTTF-TTC			160 T PRESS - II TOP FACE	??

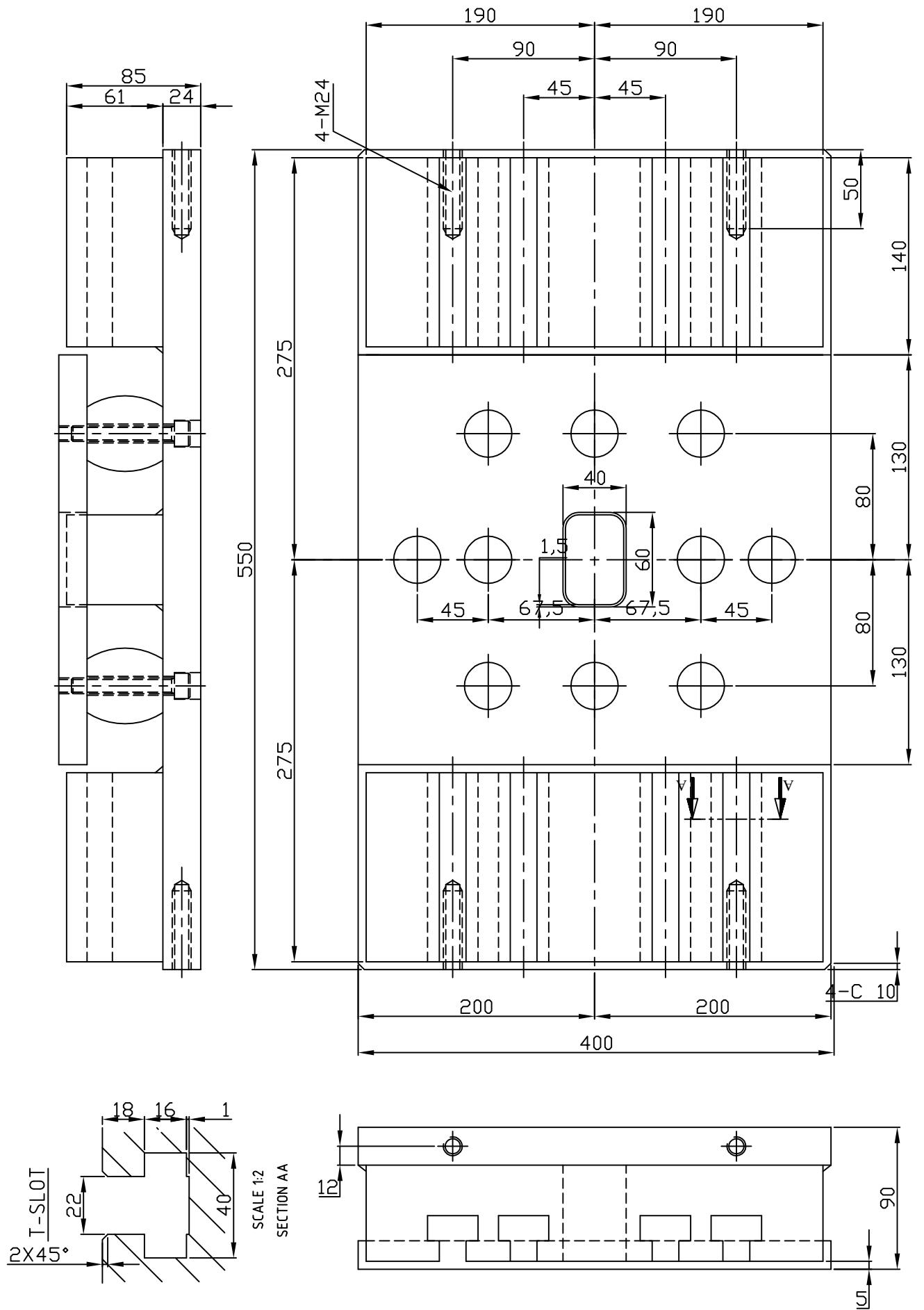


CLAMPING SCREW HOLES TO BE AS PER YOUR STD.

DRN	03 07 24	ANISH K		
CHD				
APPR				
PDTD'03-04 NTTF-TTC			100 T PRESS - IV BOTTOM FACE	??



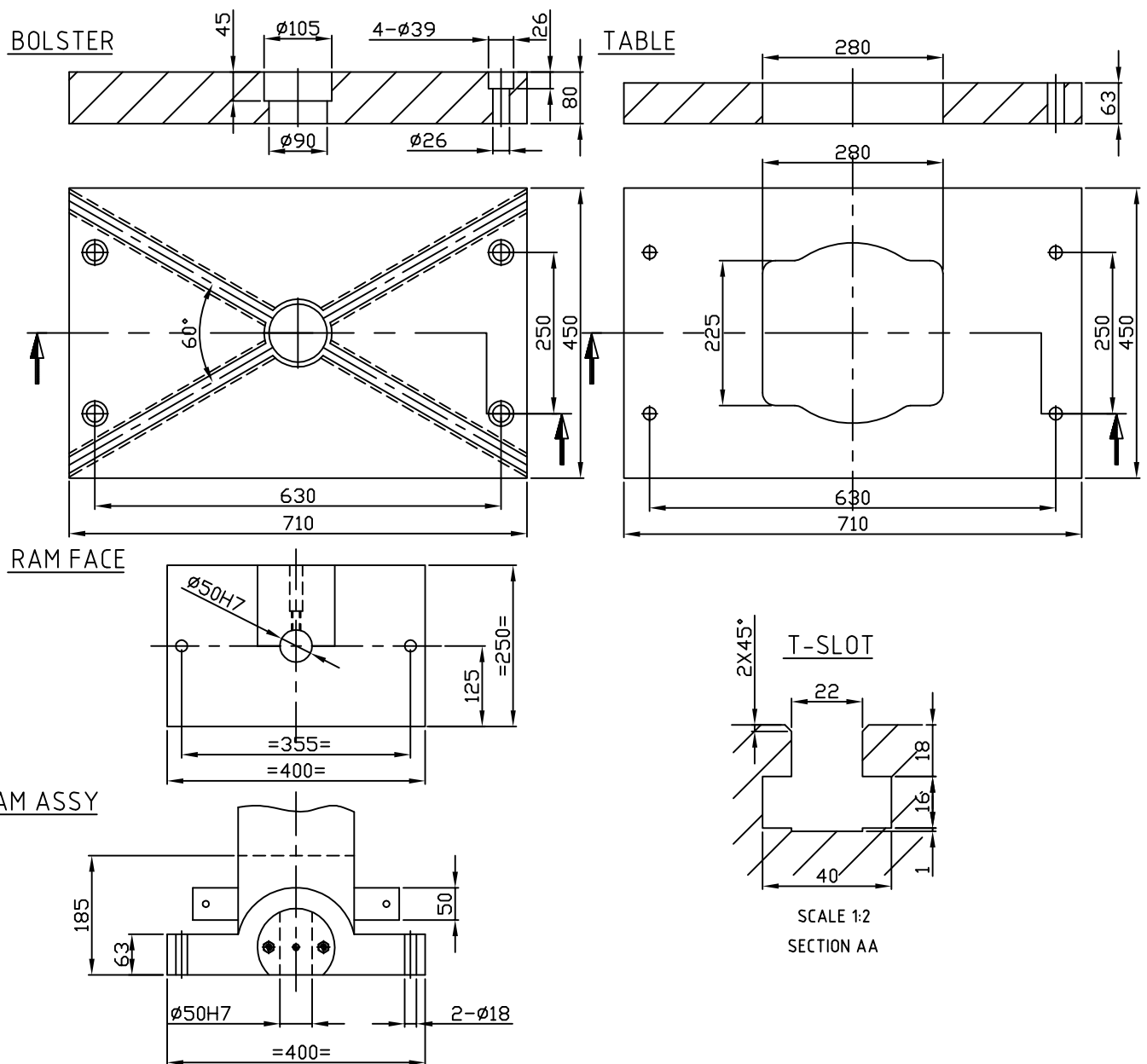
DRN	03 07 24	ANISH K		
CHD				
APPR				
PDTD'03-04 NTTF-TTC			63 T PRESS - IV & V BOTTOM FACE	??



DRN	03 07 24	ANISH K		
CHD				
APPR				
PDTD'03-04 NTTF-TTC			100 TON I TOP	??

SPECIFICATION

1. CAPACITY AT 20° CRANK ANGLE BEFORE BDC STANDARD STROKE	: 400 KN
2. STROKE ADJUSTABLE	: 8-80 mm
3. DEPTH OF THROAT	: 224 mm
4. SHUT HEIGHT FROM TABLE SURFACE (RAM DOWN ADJ. UP)	: 355 mm
5. RAM ADJUSTMENT	: 63 mm
6. RAM FACE FBX LR	: 250 x 400 mm
7. HEIGHT OF TABLE SURFACE FROM FLOOR	: 800 mm
8. DISTANCE BETWEEN UPRIGHTS	: 355 mm
9. STROKE/MIN	: 100 Spm

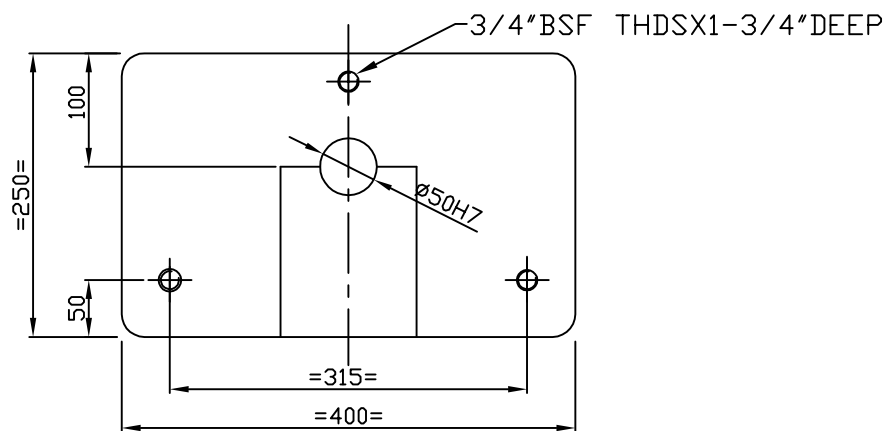
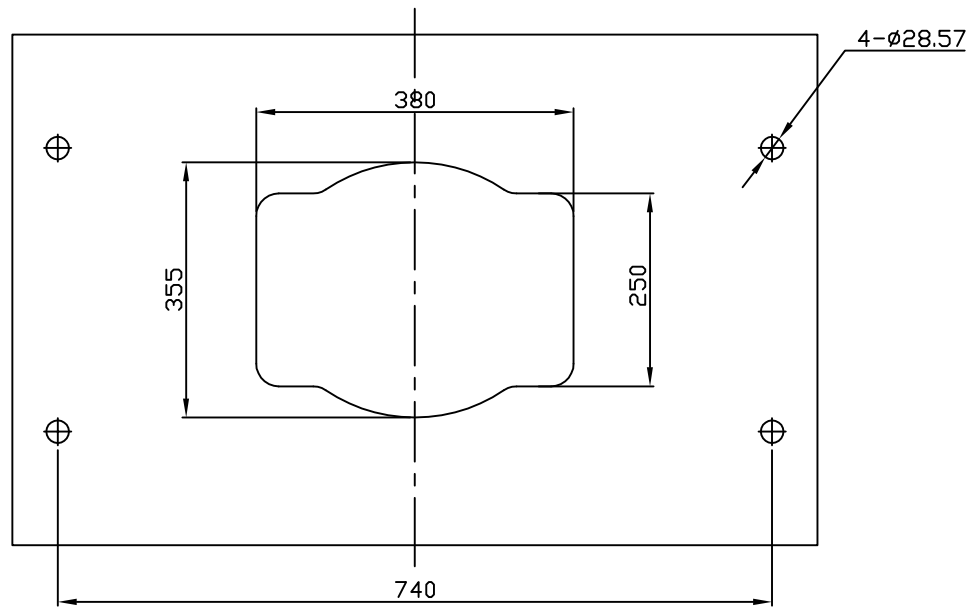
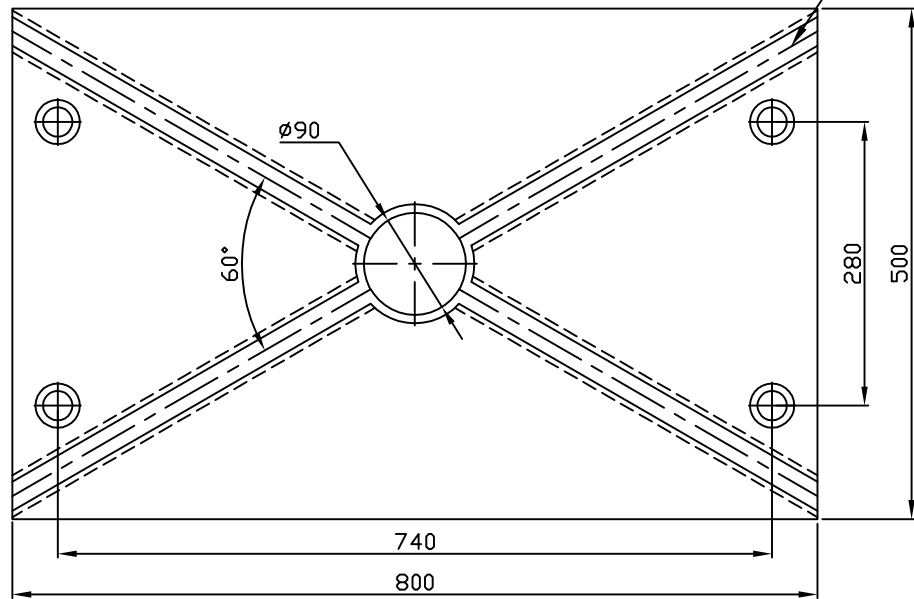


DRN	03 07 24	ANISH K
CHD		
APPR		

PDTD'03-04
NTTF-TTC

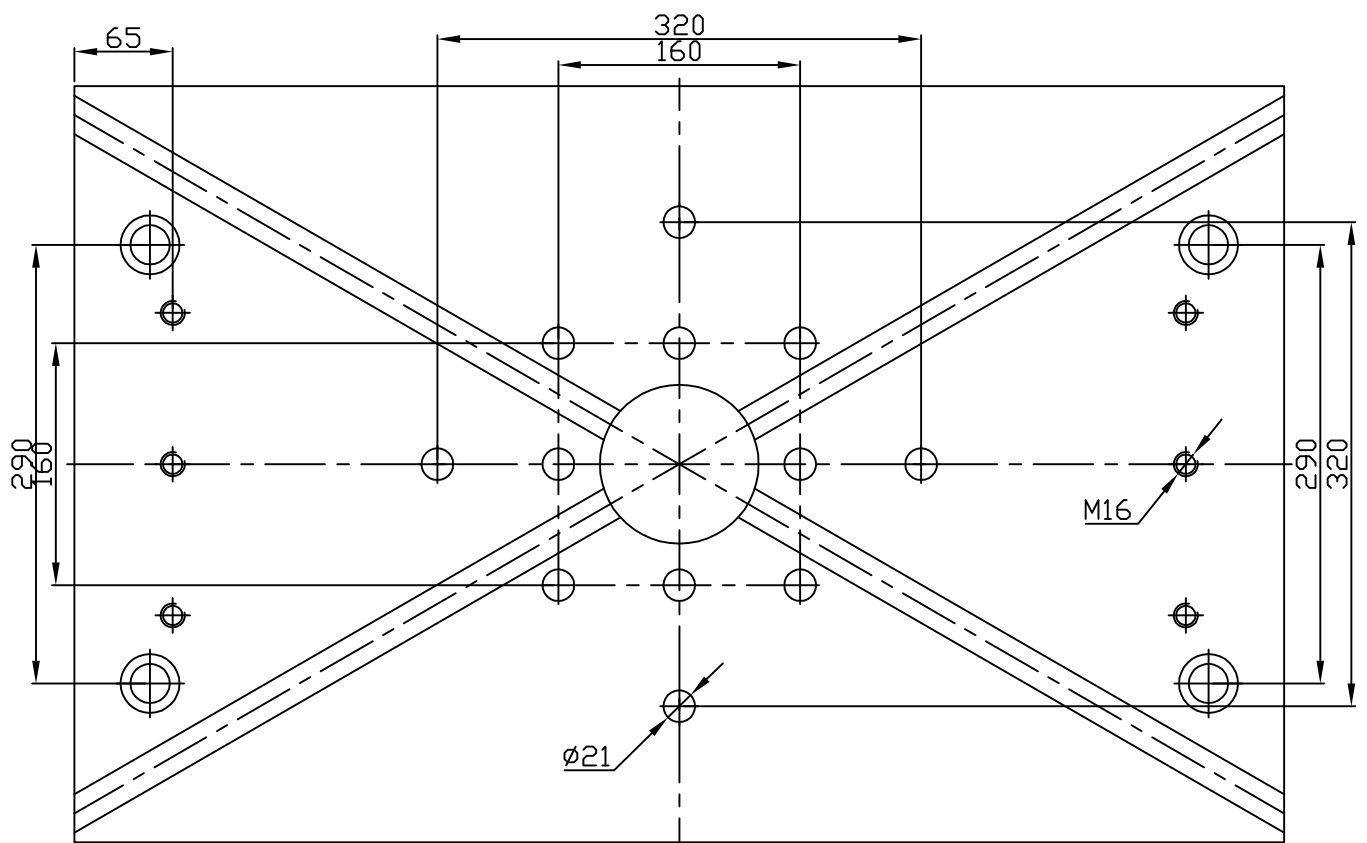
GODREJ 400 KN PRESS
NON GEARED

05-01-04-03-3



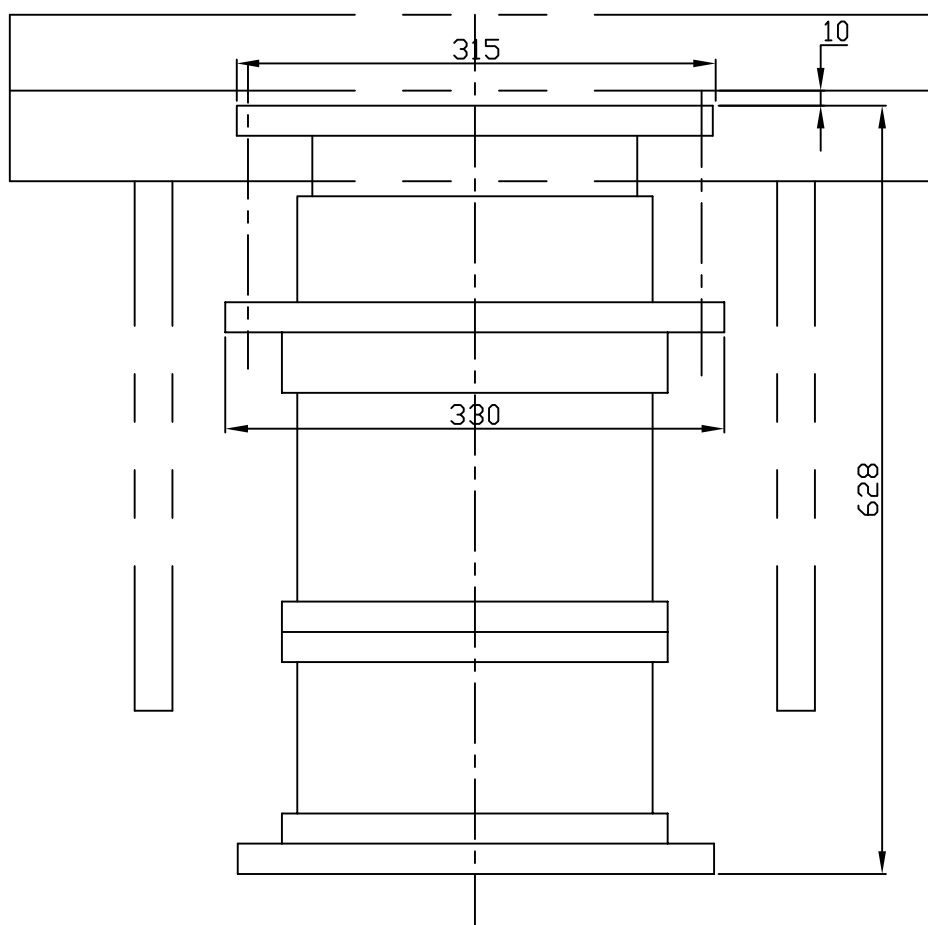
DRN	03 07 24	ANISH K	
CHD			
APPR			
PDTD'03-04 NTTF-TTC			BOLSTER PLATE GODREJ 63 TONNES I GEARED
			05-01-05-05-2

Technical drawing of a mechanical part, likely a shaft or pulley, showing a cross-section. The drawing includes a central hole with a diameter of 105 mm, a total length of 800 mm, and a thickness of 85 mm. The part is hatched to indicate a cross-section.



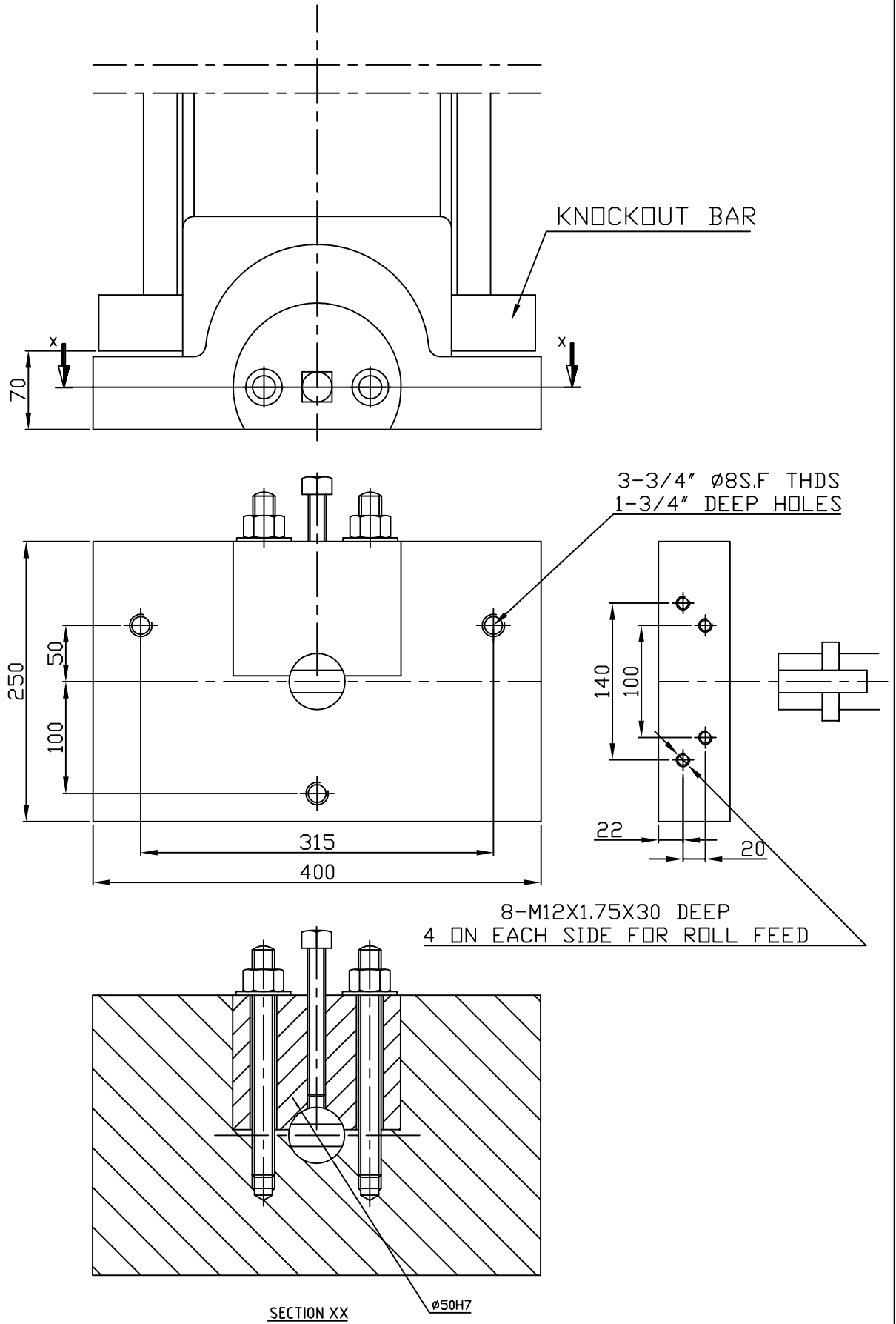
DRN	03 07 24	ANISH K	
CHD			
APPR			
PDTD'03-04 NTTT-TTC			GODREJ 63 TON (GEARED) WITH DIE CUSHION
			05-01-05-05-5

DIE CUSHION FOR 63T GODREJ PRESS.

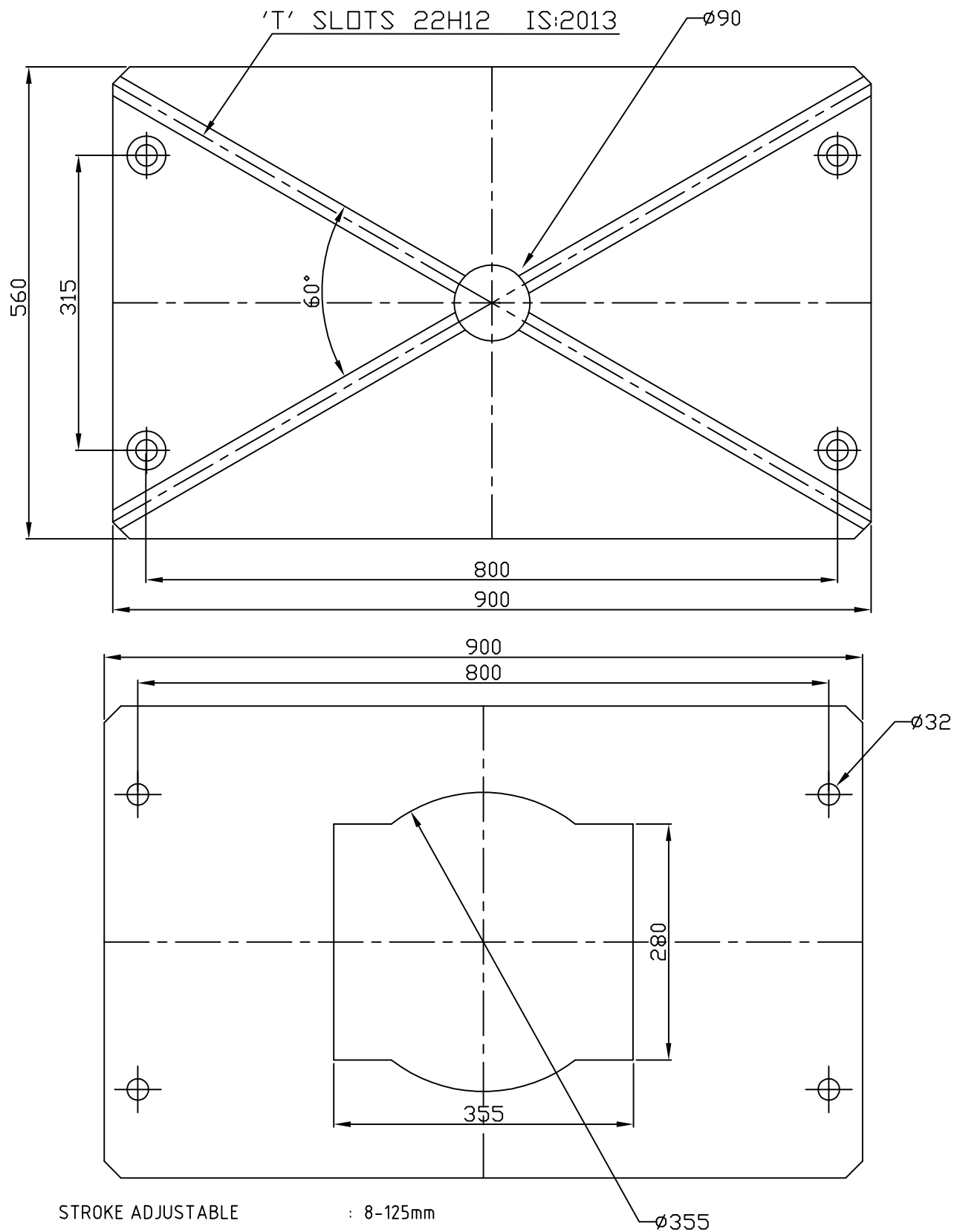


DIE CUSHION MODEL	: A3-8X70S
DIE CUSHION CAPACITY AT 7 Atms	: 8T
STROKE OF DIE CUSHION	: 70mm
DIAMETER OF PRESSURE PLATE	: 315mm
CLEARANCE BETWEEN BOLSTER & Pr. PLATE	: 10mm
CYLINDER DIAMETER	: 224mm
NO OF CYLINDERS	: 3
SIZE OF AIR CONNECTION	: 1-1/4" BSP TAPER
MAXIMUM WORKING PRESSURE	: 8kg/cm ²
WEIGHT	: 162kg
TRANSPER PIN	: $\phi 20$

DRN	03 07 24	ANISH K		
CHD				
APPR				
PDTD'03-04 NTTF-TTC			DIE CUSHION FOR 63 T GODREJ PRESS	05-01-05-05-4

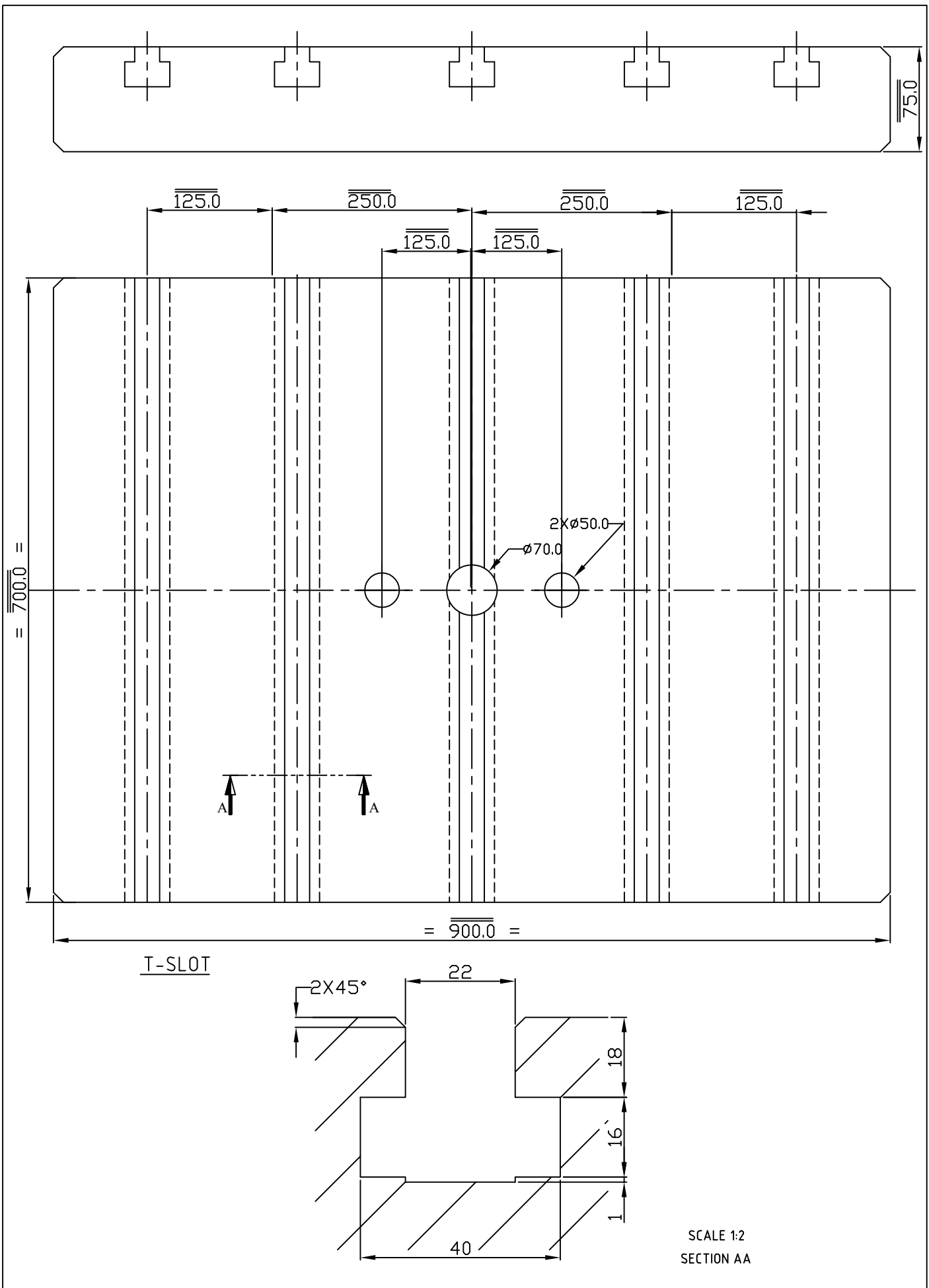


DRN	03 07 24	ANISH K	
CHD			
APPR			
PDTD'03-04 NTTF-TTC			GODREJ 63 TON (GEARED)
			05-01-05-05-3

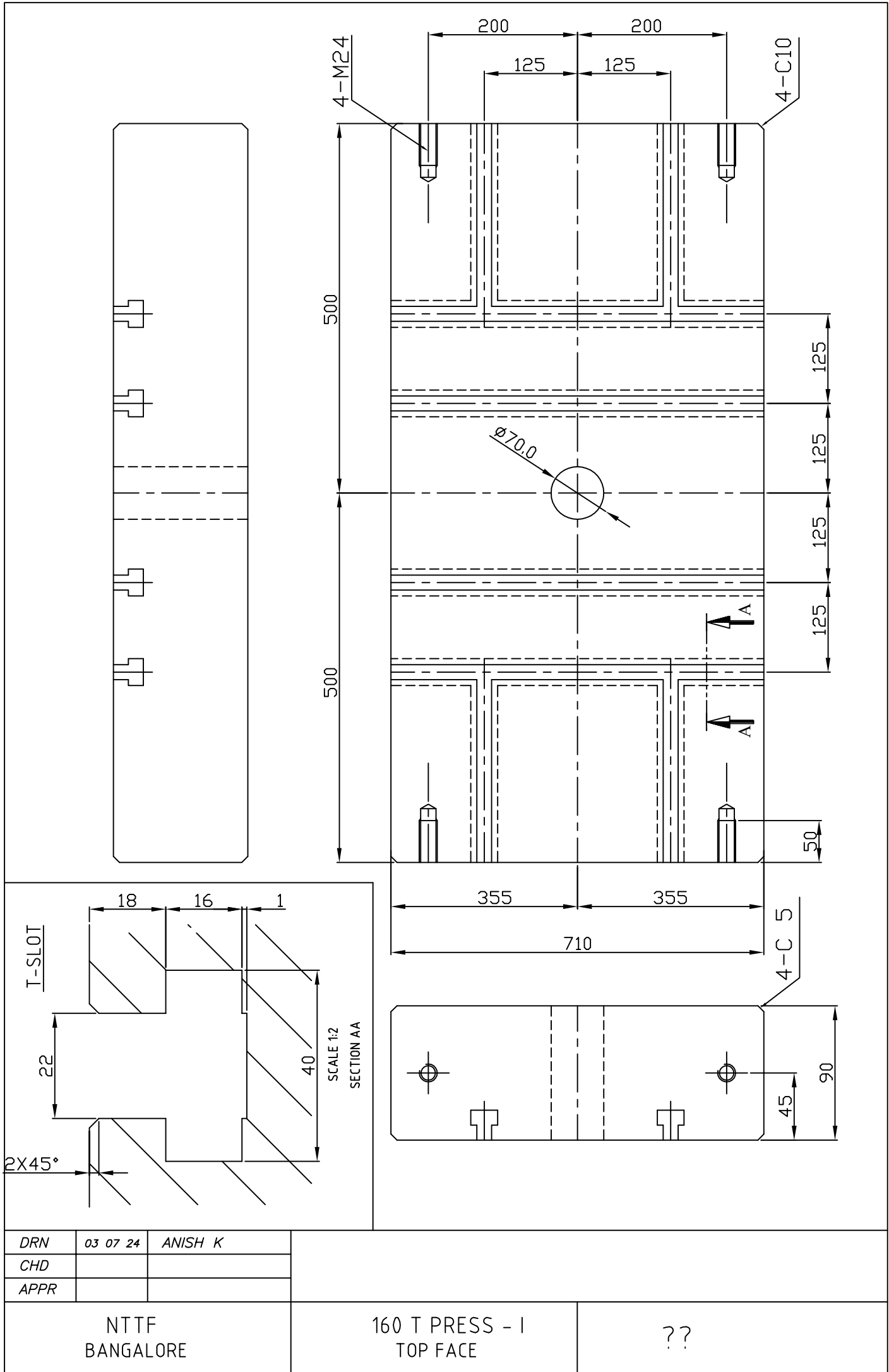


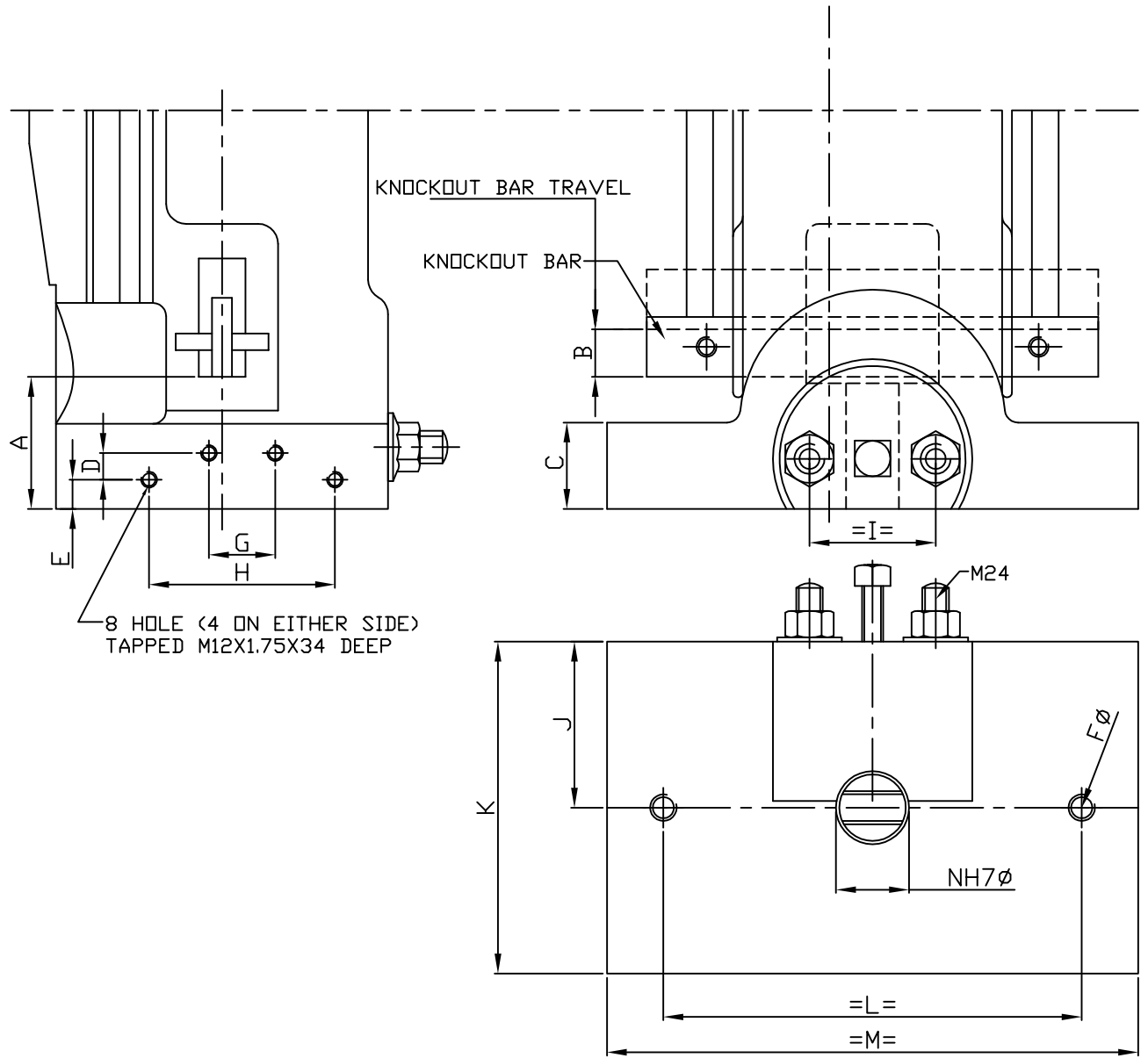
STROKE ADJUSTABLE	: 8-125mm
DEPTH OF THROAT	: 280mm
SHUT HEIGHT	: 355mm. (AT 125mm STROKE)
RAM ADJUSTMENT	: 80mm
STROKES PER MIN.	: 40
SIZE OF TABLE	: 560X900mm
BOLSTER THICKNESS	: 100mm
DISTANCE BETWEEN UPRIGHTS	: 530mm

DRN	03 07 24	ANISH K	
CHD			
APPR			
PDTD'03-04 NTTF-TTC			GODREJ 100 T BOTTEM FACE
			05-01-07-03-2



DRN	03 07 24	ANISH K		
CHD				
APPR				
PDTD'03-04 NTTF-TTC			GODREJ 315 T PRESS TOP FACE	05-01-09-05-4



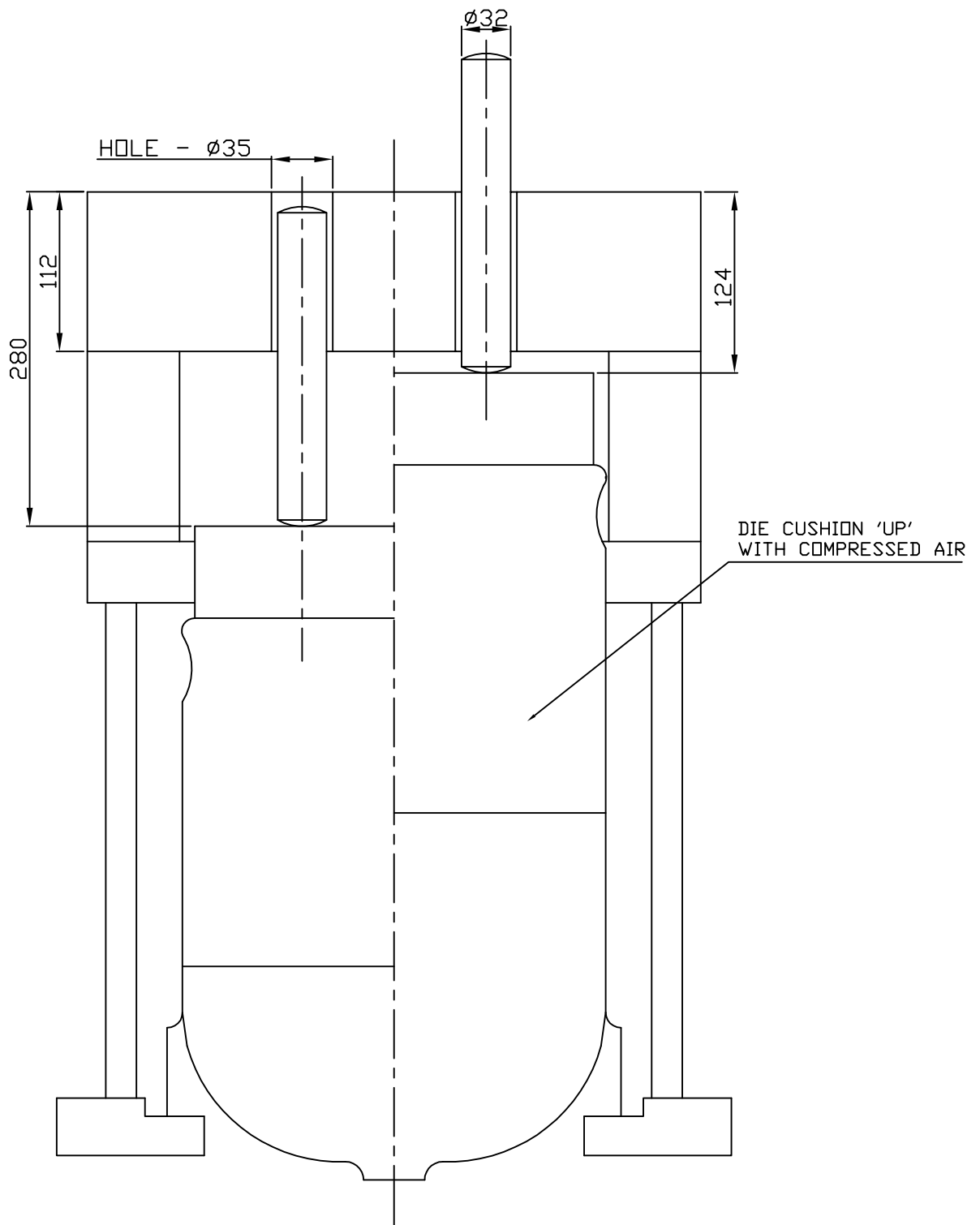


DRN	03 07 24	ANISH K
CHD		
APPR		

PDTD'03-04
NTTF-TTC

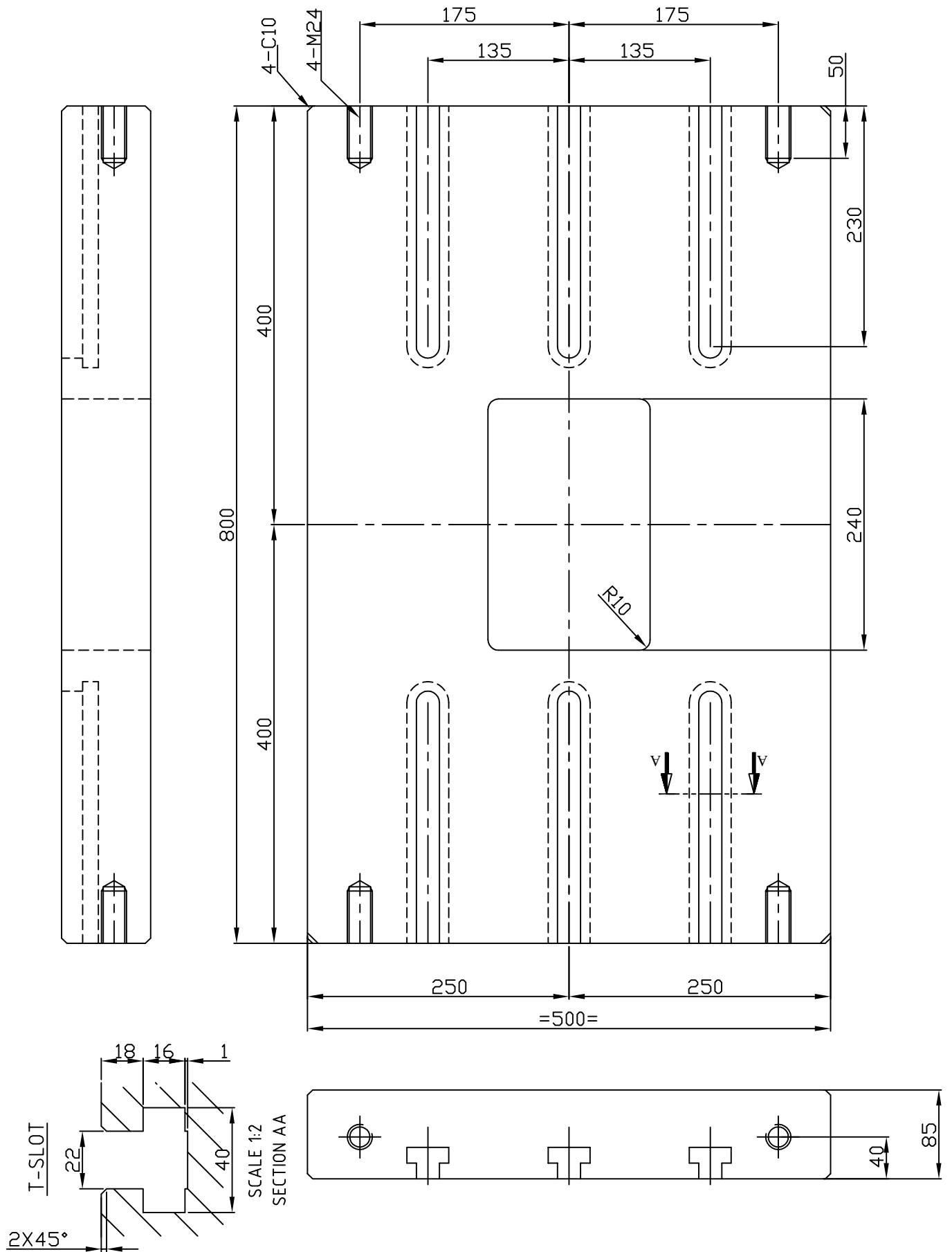
SIDE FACE OF
GODREJ 100 TON II

05-01-07-03-3

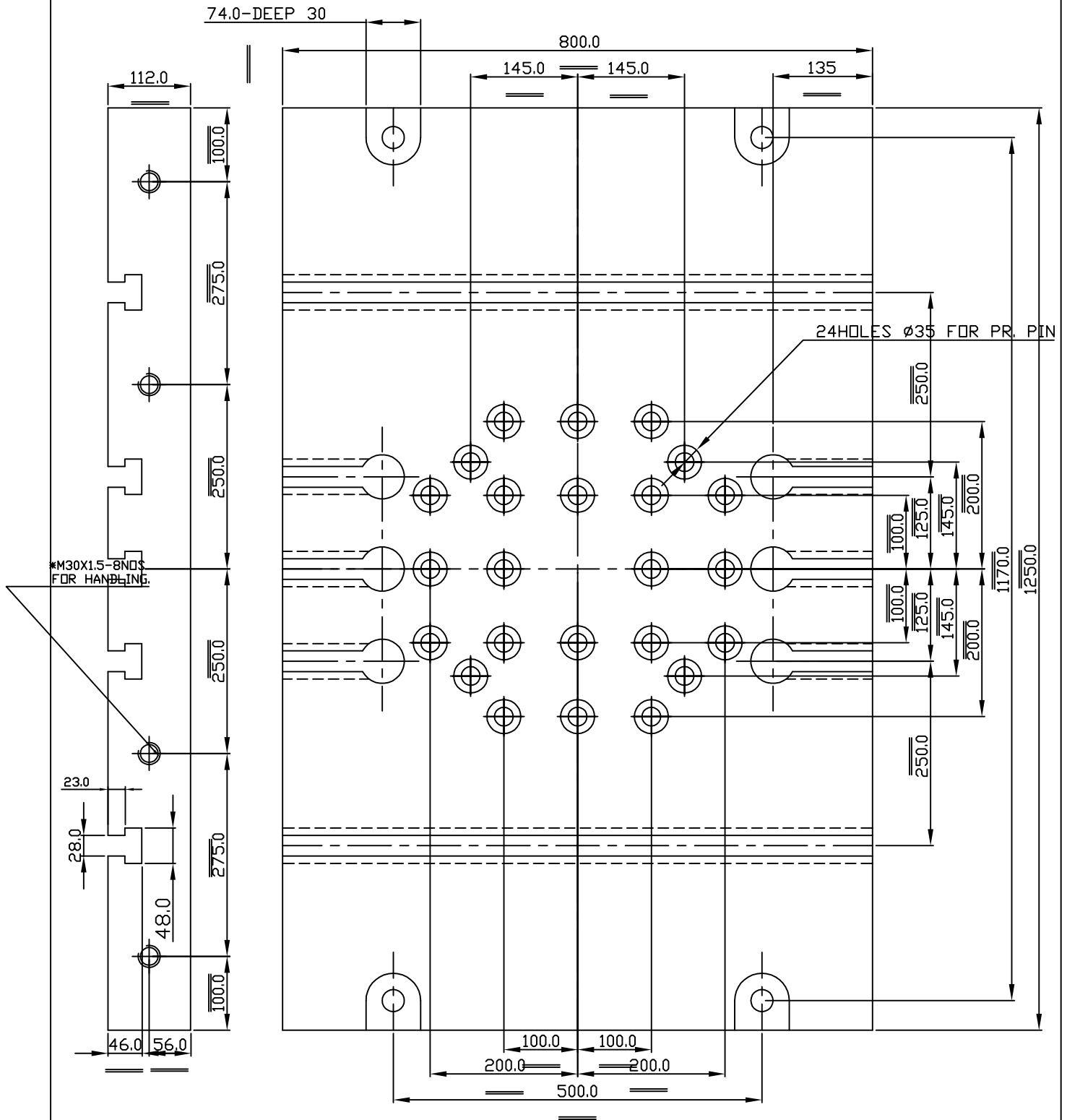


DIE CUSHION STROKE - 156 mm(315 TON)

DRN	03 07 24	ANISH K		
CHD				
APPR				
PDTD'03-04 NTTF-TTC			GODREJ 315 T PRESS DIE CUSHION	05-01-09-05-5



DRN	03 07 24	ANISH K		
CHD				
APPR				
NTTF BANGALORE			63 T PRESS - III BOTTOM FACE	??

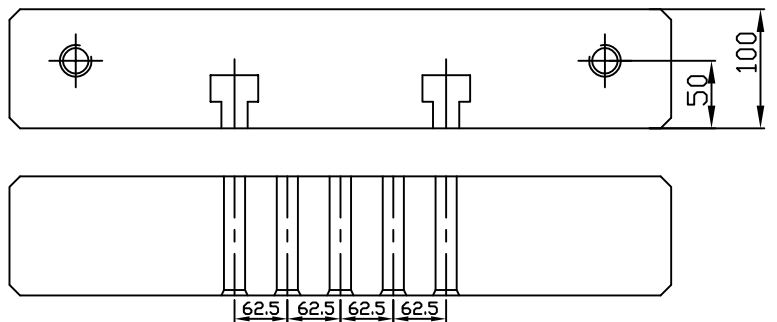
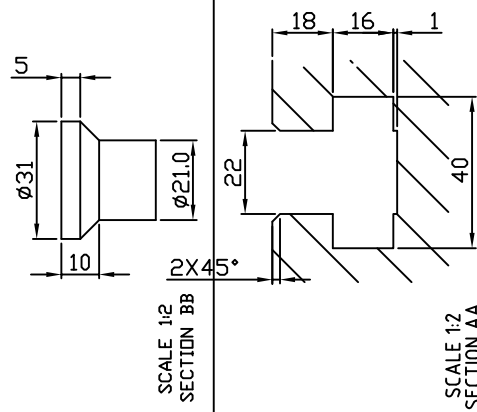
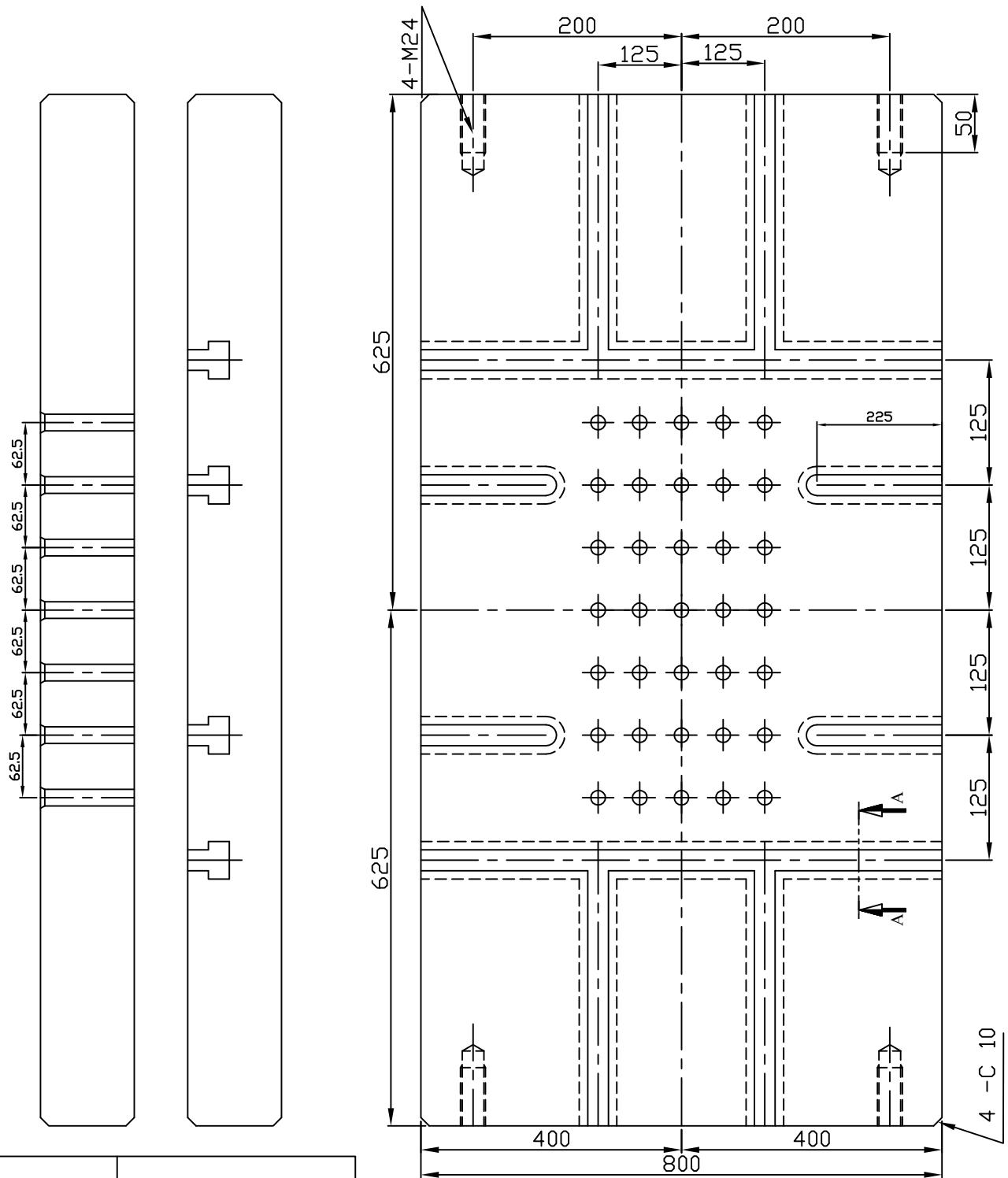


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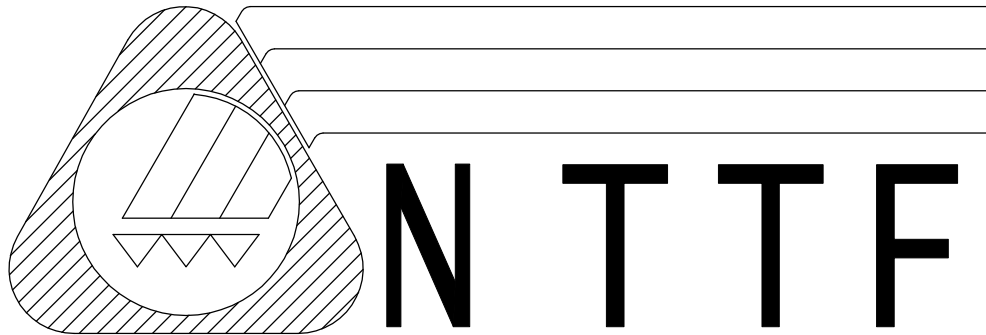


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NTTF BANGALORE			160 T PRESS - I BOTTOM FACE
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UNBRAKO

SCREWS &

DOWELS



**COMPILED BY SRIVINAY
PGDTD 2009,
NTTF BANGALORE**

THREAD CONVERSION CHARTS

DIAMETER/THREAD PITCH COMPARISON

INCH SERIES			METRIC			
Size	Dia. (In.)	TPI	Size	Dia.(In.)	Pitch (mm)	TPI (Approx)
#0	0.060	80				
			M1.6	0.063	0.35	74
#1	0.073	64				
			M2	0.079	0.4	64
#2	0.096	56				
			M2.5	0.098	0.45	56
#3	0.099	48				
#4	0.112	40				
			M3	0.118	0.5	51
#5	0.125	40				
#6	0.138	32				
			M4	0.157	0.7	36
#8	0.164	32				
#10	0.190	24				
			M5	0.196	0.8	32
			M6	0.236	1.00	25
1/4	0.250	20				
5/16	0.312	18				
			M8	0.315	1.25	20
3/8	0.375	16				
			M10	0.393	1.5	17
7/16	0.437	14				
			M12	0.472	1.75	14.5
1/2	0.500	13				
			M14	0.551	2	12.5
5/8	0.625	11				
			M16	0.63	2	12.5
3/4	0.750	10				
			M20	0.787	2.5	10
7/8	0.875	9				
			M24	0.945	3	8.5
1	1.000	8				
			M27	1.063	3	8.5

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UNBRAKO Metric Fasteners

UNBRAKO Metric Fasteners are the strongest off-the-shelf threaded fasteners you can buy. Their exclusive design features and closely controlled manufacturing processes insure the dimensional accuracy, strength and fatigue resistance needed for reliability in today's advanced technology. They are manufactured with the same methods and features as their inch-series counterpart.

Strength

UNBRAKO metric socket head cap screws are made into property class 12.9 with a minimum ultimate tensile strength of 1300 or 1250 MPa depending on screw diameter. Precision in manufacturing and careful control in stress areas insure strength in such critical areas as heads, sockets, threads, fillets, and bearing areas.

When you purchase UNBRAKO metric socket screw products, you can be sure that they meet or exceed the strength levels of all current standards, including the three most common-ANSI, ISO and DIN. Unbrako is represented on several ASME, ANSI, ASTM and ISO committees.

- ANSI (American National Standards Institute) documents are published by ASME (The American Society of Mechanical Engineers) and are familiar to almost all users of socket screw products in the U.S.A.
- ASTM (American Society for Testing and Materials). Many ANSI documents list dimensional information but refer to ASTM specifications for materials, mechanical properties, and test criteria.

- ISO (International Standards Organization) is a standards group comprising 70 member nations. Its objective is to provide standards that will be completely universal and common to all countries subscribing.

- DIN (Deutsche Industries Normen) is the German standards group.

NOTE: The proper tightening of threaded fasteners can have a significant effect on their performance.

A WARNING TO METRIC FASTENER USERS

Metric socket cap screws are NOT sold in a single strength level like U.S. inch socket screws.

Property Class	General Material	Strength Level, UTS min. MPa (KSI)
	International Standards Organization, ISO	
Property Class 8.8	Carbon Steel	800 (116) < M16 830 (120) ≥ M16
Property Class 10.9 Property Class 12.9	Alloy Steel Alloy Steel	1040 (151) 1220 (177)
USA Standards ASTM A574M	Alloy Steel	1220 (177)
Unbrako Standards ASTM A574M	Alloy Steel	1300 (189) ≤ M16 1250 (181) > M16

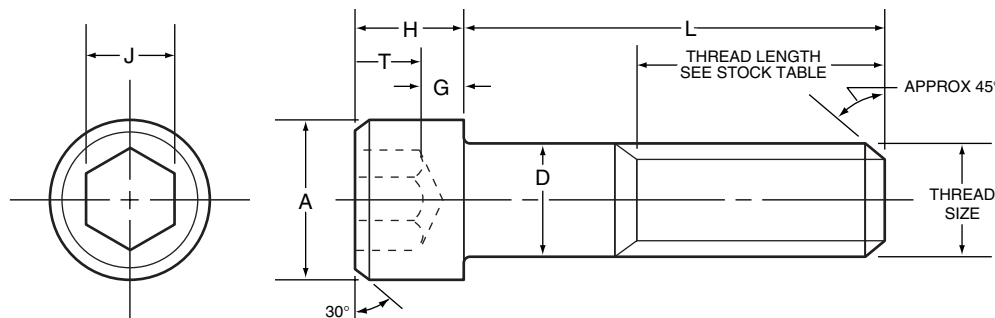
STANDARDS

The use of metric fasteners in the worldwide market has led to the creation of many standards. These standards specify the fastener requirements: dimensions, material, strength levels, inspection, etc. Different standards are the responsibility of various organizations and are not always identical. Unbrako supplies metric fasteners for maximum interchangeability with all standards. This Engineering Guide was published with the most current values, which are however subject to change by any standards organization at any time.

METRIC SOCKET HEAD CAP SCREWS

Dimensions

Threads: ANSI B1.13M, ISO 261, ISO 262 (coarse series only)
Property Class: 12.9-ISO 898/1



NOTES

- Material:** ASTM A574M, DIN ENISO4762-alloy steel
- Hardness:** Rc 38-43
- Tensile Stress:** 1300 MPa thru M16 size.
1250 MPa over M16 size.
- Yield Stress:** 1170 MPa thru M16 size.
1125 MPa over M16 size.
- Thread Class:** 4g 6g

LENGTH TOLERANCE

nominal screw length	nominal screw diameter		
	M1.6 thru M10	M12 thru M20	over 20
	tolerance on lgth., mm		
Up to 16 mm, incl.	±0.3	±0.3	—
Over 16 to 50 mm, incl.	±0.4	±0.4	±0.7
Over 50 to 120 mm, incl.	±0.7	±1.0	±1.5
Over 120 to 200 mm, incl.	±1.0	±1.5	±2.0
Over 200 mm	±2.0	±2.5	±3.0

DIMENSIONS

MECHANICAL PROPERTIES

APPLICATION DATA

thread size nom.	pitch	A	D	H	J	G	T	UTS min. MPa	tensile strength min.		single shear strength of body min.		recommended ** seating torque plain finish	
		max.	max.	max.	nom.	min.	min.		kN	lbs.	kN	lbs.	N-m	in-lbs.
M1.6	0.35	3.0	1.6	1.6	1.5	0.54	0.80	1300	1.65	370	1.57	352.5	0.29	2.6
M2	0.40	3.8	2.0	2.0	1.5	0.68	1.0	1300	2.69	605	2.45	550	0.60	5.3
M2.5	0.45	4.5	2.5	2.5	2.0	0.85	1.25	1300	4.41	990	3.83	860	1.21	11
M3	0.5	5.5	3.0	3.0	2.5	1.02	1.5	1300	6.54	1,470	5.5	1240	2.1	19
M4	0.7	7.0	4.0	4.0	3.0	1.52	2.0	1300	11.4	2,560	9.8	2,205	4.6	41
M5	0.8	8.5	5.0	5.0	4.0	1.90	2.5	1300	18.5	4,160	15.3	3,445	9.5	85
M6	1.0	10.0	6.0	6.0	5.0	2.28	3.0	1300	26.1	5,870	22.05	4,960	16	140
M8	1.25	13.0	8.0	8.0	6.0	3.2	4.0	1300	47.6	10,700	39.2	8,800	39	350
M10	1.5	16.0	10.0	10.0	8.0	4.0	5.0	1300	75.4	17,000	61	13,750	77	680
M12	1.75	18.0	12.0	12.0	10.0	4.8	6.0	1300	110	24,700	88	19,850	135	1,200
*(M14)	2.0	21.0	14.0	14.0	12.0	5.6	7.0	1300	150	33,700	120	27,000	215	1,900
M16	2.0	24.0	16.0	16.0	14.0	6.4	8.0	1300	204	45,900	157	35,250	330	2,900
M20	2.5	30.0	20.0	20.0	17.0	8.0	10.0	1250	306	68,800	235.5	53,000	650	5,750
M24	3.0	36.0	24.0	24.0	19.0	9.6	12.0	1250	441	99,100	339	76,500	1100	9,700
*M30	3.5	45.0	30.0	30.0	22.0	12.0	15.0	1250	701	158,000	530	119,000	2250	19,900
*M36	4.0	54.0	36.0	36.0	27.0	14.4	18.0	1250	1020	229,000	756	171,500	3850	34,100
*M42	4.5	63.0	42.0	42.0	32.0	16.8	21.0	1250	1400	315,000	1040	233,500	6270	55,580
*M48	5.0	72.0	48.0	48.0	36.0	19.2	24.0	1250	1840	413,000	1355	305,000	8560	75,800

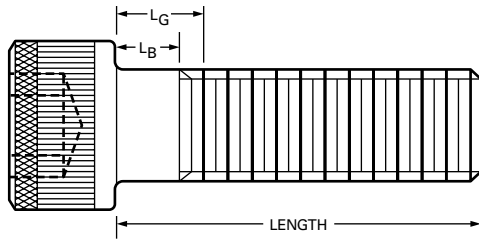
All dimensions in millimeters.

Sizes in brackets not preferred for new designs.

*Non-stock diameter.

**Torque calculated in accordance with VDI 2230, "Systematic Calculation of High Duty Bolted Joints," to induce approximately 800 MPa stress in screw threads. Torque values listed are for plain screws. (See Note, page 1.)

SOCKET HEAD CAP SCREWS ■ Metric ■ Body and Grip Lengths



L_G is the maximum grip length and is the distance from the bearing surface to the first complete thread.

L_B is the minimum body length and is the length of the unthreaded cylindrical portion of the shank.

BODY and GRIP LENGTHS

BODY AND GRIP LENGTH DIMENSIONS FOR METRIC SOCKET HEAD CAP SCREWS

Nominal Size	M1.6		M2		M2.5		M3		M4		M5		M6		M8		M10		M12		M14		M16		M20		M24	
Nominal Length	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B
20	4.8	3.0	4.0	2.0																								
25	9.8	8.0	9.0	7.0	8.0	5.7	7.0	4.5																				
30	14.8	13.0	14.0	12.0	13.0	10.7	12.0	9.5	10.0	6.5																		
35	...	...	19.0	17.0	18.0	15.7	17.0	14.5	15.0	11.5	13.0	9.0	11.0	6.0														
40	...	...	24.0	22.0	23.0	20.7	22.0	19.5	20.0	16.5	18.0	14.0	16.0	11.0														
45	...	...	...	...	28.0	25.7	27.0	24.5	25.0	21.5	23.0	19.0	21.0	16.0	17.0	10.7												
50	...	...	...	...	33.0	30.7	32.0	29.5	30.0	26.5	28.0	24.0	26.0	21.0	22.0	15.7	18.0	10.5										
55	...	...	...	...	...	...	37.0	34.5	35.0	31.5	33.0	29.0	31.0	26.0	27.0	20.7	23.0	15.5										
60	...	...	...	...	...	...	42.0	39.5	40.0	36.5	38.0	34.0	36.0	31.0	32.0	25.7	28.0	20.5	24.0	15.2								
65	...	...	...	...	...	...	47.0	44.5	45.0	41.5	43.0	39.0	41.0	36.0	37.0	30.7	33.0	25.5	29.0	20.2	25.0	15.0						
70	...	...	...	...	...	...	...	...	50.0	46.5	48.0	44.0	46.0	41.0	42.0	35.7	38.0	30.5	34.0	25.2	30.0	20.0	26.0	16.0				
80	...	...	...	...	...	...	...	...	60.0	56.5	58.0	54.0	56.0	51.0	52.0	45.7	48.0	40.5	44.0	35.2	40.0	30.0	36.0	26.0				
90	...	...	...	...	...	...	...	...	...	...	68.0	64.0	66.0	61.0	62.0	55.7	58.0	50.5	54.0	45.2	50.0	40.0	46.0	36.0	38.0	25.5		
100	...	...	...	...	...	...	...	...	...	...	78.0	74.0	76.0	71.0	72.0	65.7	68.0	60.5	64.0	55.2	60.0	50.0	56.0	46.0	48.0	35.5	40.0	25.0
110	...	...	...	...	...	...	...	...	...	...	...	...	86.0	81.0	82.0	75.7	78.0	70.5	74.0	65.2	70.0	60.0	66.0	56.0	58.0	45.5	50.0	35.0
120	...	...	...	...	...	...	...	...	...	...	...	...	96.0	91.0	92.0	85.7	88.0	80.5	84.0	75.2	80.0	70.0	76.0	66.0	68.0	55.5	60.0	45.0
130	...	...	...	...	...	...	...	...	...	...	...	...	...	...	102.0	95.7	98.0	90.5	94.0	85.2	90.0	80.0	86.0	76.0	78.0	65.5	70.0	55.0
140	...	...	...	...	...	...	...	...	...	...	...	...	...	...	112.0	105.7	108.0	100.5	104.0	95.2	100.0	90.0	96.0	86.0	88.0	75.5	80.0	65.0
150	...	...	...	...	...	...	...	...	...	...	...	...	...	...	122.0	115.7	118.0	110.5	114.0	105.2	110.0	100.0	106.0	96.0	98.0	85.5	90.0	75.0
160	...	...	...	...	...	...	...	...	...	...	...	...	...	...	132.0	125.7	128.0	120.5	124.0	115.2	120.0	110.0	116.0	106.0	108.0	95.5	100.0	85.0
180	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	148.0	140.5	144.0	135.2	140.0	130.0	136.0	126.0	128.0	115.5	120.0	105.0
200	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	168.0	160.5	164.0	155.2	160.0	150.0	156.0	146.0	148.0	135.5	140.0	125.0
220	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	184.0	175.2	180.0	170.0	176.0	166.0	168.0	155.5	160.0	145.0
240	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	204.0	195.2	200.0	190.0	196.0	186.0	188.0	175.5	180.0	165.0
260	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	220.0	210.0	216.0	206.0	208.0	195.5	200.0	185.0
300	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	256.0	246.0	248.0	235.5	240.0	225.0

SOCKET HEAD CAP SCREWS (METRIC SERIES)

PER ASME/ANSI B18.3.1M-1986

METRIC SOCKET FLAT HEAD CAP SCREWS

Dimensions

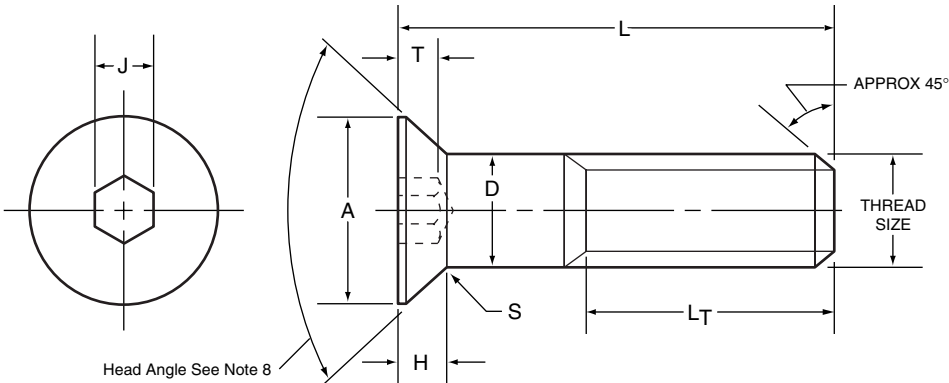
Threads: ANSI B1.13M, ISO 262 (coarse series only)

Applicable or Similar Specification: DIN ENISO10642

General Note: Flat, countersunk head cap screws and button head cap screws are designed and recommended for moderate fastening applications: machine guards, hinges, covers, etc. They are not suggested for use in critical high strength applications where socket head cap screws should be used.

NOTES

- 1. Material:** ASTM F835M
- 2. Dimensions:** B18.3.5M
- 3. Property Class:** 12.9
- 4. Hardness:** Rc 38-43 (alloy steel)
- 5. Tensile Stress:** 1040MPa
- 6. Shear Stress:** 630 MPa
- 7. Yield Stress:** 945 MPa
- 8. Sizes:** For sizes up to and including M20, head angle shall be 92°/90°. For larger sizes head angle shall be 62°/60°.
- 9. Thread Class:** 4g 6g



LENGTH TOLERANCE

nominal screw length	nominal screw diameter	
	M3 thru M24	
	tolerance on lgth., mm	
Up to 16 mm, incl.	±0.3	
Over 16 to 60 mm, incl.	±0.5	
Over 60 mm	±0.8	

DIMENSIONS

APPLICATION DATA

nom. thread size	pitch	A	D	H	T	S	L _T	J	recommended seating torque**	
		max.***	max.	ref.	min.	ref.	min.	nom.	plain	
									N-m	in-lbs.
M3	0.5	6.72	3	1.7	1.10	0.50	18	2	1.2	11
M4	0.7	8.96	4	2.3	1.55	0.70	20	2.5	2.8	25
M5	0.8	11.20	5	2.8	2.05	0.70	22	3	5.5	50
M6	1.0	13.44	6	3.3	2.25	0.85	24	4	9.5	85
M8	1.25	17.92	8	4.4	3.20	1.20	28	5	24	210
M10	1.50	22.40	10	5.5	3.80	1.50	32	6	47	415
M12	1.75	26.88	12	6.5	4.35	1.85	36	8	82	725
M16	2.00	33.60	16	7.5	4.89	1.85	44	10	205	1800
M20	2.50	40.32	20	8.5	5.45	1.85	52	12	400	3550
*M24	3.00	40.42	24	14.0	10.15	2.20	60	14	640	5650

All dimensions in millimeters.

*Non-stock Diameter

**Torque calculated to induce 420 MPa in the screw threads.

Torque values are for plain screws. (See Note, page 1.)

***Maximum to theoretical sharp corner

METRIC SOCKET BUTTON HEAD CAP SCREWS

Dimensions

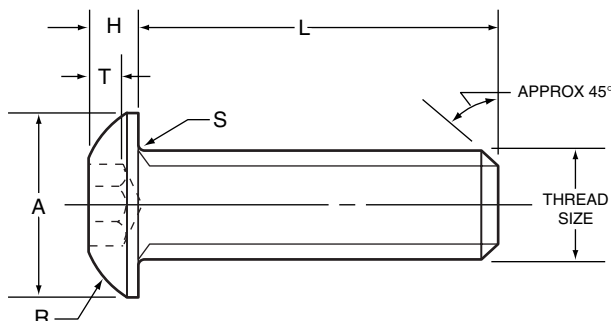
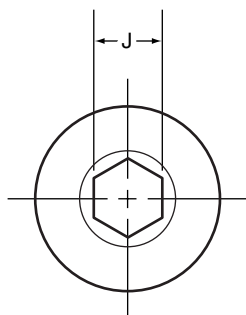
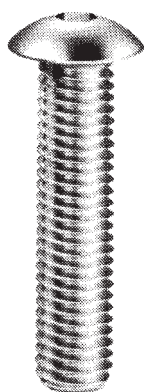
Threads: ANSI B1.13M, ISO 262(coarse series only)

Similar Specifications: ISO 7380

General Note: Flat, countersunk head cap screws and button head cap screws are designed and recommended for moderate fastening applications: machine guards, hinges, covers, etc. They are not suggested for use in critical high strength applications where socket head cap screws should be used.

NOTES

1. **Material:** ASTM F835M
2. **Dimensions:** ANSI B18.3.4M
3. **Property Class:** 12.9
4. **Hardness:** Rc 38-43
5. **Tensile Stress:** 1040 MPa
6. **Shear Stress:** 630 MPa
7. **Yield Stress:** 945 MPa
8. Bearing surface of head square with body within 2°.
9. **Thread Class:** 4g 6g



LENGTH TOLERANCE

nominal screw length	nominal screw diameter
	M3 thru M16
	tolerance on lgth., mm
Up to 16 mm, incl.	±0.3
Over 16 to 60 mm, incl.	±0.5
Over 60 mm	±0.8

DIMENSIONS

APPLICATION DATA

nom. thread size	pitch	A	H	T	R	S	J	recommended seating torque**	
		max.	max.	min.	ref.	ref.	nom.	plain	
								N-m	in-lbs.
M3	0.5	5.70	1.65	1.05	2.95	.35	2.0	1.2	11
M4	0.7	7.60	2.20	1.35	4.10	.35	2.5	2.8	25
M5	0.8	9.50	2.75	1.92	5.20	.45	3.0	5.5	50
M6	1.0	10.50	3.30	2.08	5.60	.45	4.0	9.5	85
M8	1.285	14.00	4.40	2.75	7.50	.45	5.0	24.0	210
M10	1.50	18.00	5.50	3.35	10.00	.60	6.0	47.0	415
M12	1.75	21.00	6.60	4.16	11.00	.60	8.0	82.0	725
*M16	2.0	28.00	8.60	5.20	15.00	.60	10.0	205.0	1800

All dimensions in millimeters.

*Non-stock Diameter

**Torque calculated to induce 420 MPa in the screw threads.

Torque values are for plain screws. (See Note, page 1.)

METRIC SOCKET HEAD SHOULDER SCREWS

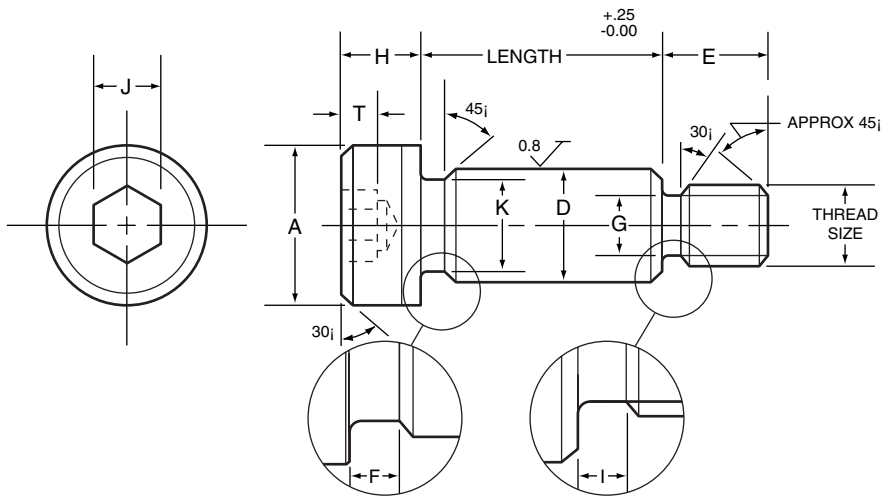
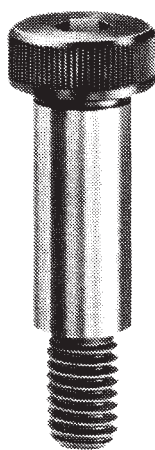
Threads: ANSI B 1.13 M, ISO 262

Similar Specifications: ANSI B18.3.3M,
ISO 7379, DIN 9841

NOTES

- 1. **Material:** ASTM A574M alloy steel
- 2. **Hardness:** Rc 36-43
- 3. **Tensile Stress:** 1100 MPa based on minimum thread neck area (G min.).
- 4. **Shear Stress:** 660 MPa
- 5. **Concentricity:** Body to head O.D. within 0.15 TIR when checked in a “V” block.

Body to thread pitch diameter within 0.1 TIR when checked at a distance of 5mm from the shoulder at the threaded end.
- Squareness, concentricity, parallelism, and bow of body to thread pitch diameter shall be within 0.05 TIR per centimeter of body length with a maximum of 0.5 when seated against the shoulder in a threaded bushing and checked on the body at a distance of 2.5 “D” from the underside of the head.
- 6. **Squareness:** The bearing surface of the head shall be perpendicular to the axis of the body within a maximum deviation of 2°.
- 7. **Thread Class:** 4g 6g



DIMENSIONS

APPLICATION DATA

nom. size	thread size	pitch	A max.	T min.	D*		K min.	H max.	G min.	F max.	I max.	E max.	J nom.	recommended seating torque**	
					max.	min.								N-m	in-lbs.
6	M5	0.8	10.00	2.4	6.0	5.982	5.42	4.50	3.68	2.5	2.40	9.75	3	7	60
8	M6	1.0	13.00	3.3	8.0	7.978	7.42	5.50	4.40	2.5	2.60	11.25	4	12	105
10	M8	1.25	16.00	4.2	10.0	9.978	9.42	7.00	6.03	2.5	2.80	13.25	5	29	255
12	M10	1.5	18.00	4.9	12.0	11.973	11.42	8.00	7.69	2.5	3.00	16.40	6	57	500
16	M12	1.75	24.00	6.6	16.0	15.973	15.42	10.00	9.35	2.5	4.00	18.40	8	100	885
20	M16	2.0	30.00	8.8	20.0	19.967	19.42	14.00	12.96	2.5	4.80	22.40	10	240	2125
24	M20	2.5	36.00	10.0	24.0	23.967	23.42	16.00	16.30	3.0	5.60	27.40	12	470	4160

All dimensions in millimeters.
*Shoulder diameter tolerance h8 (ISO R 286)

**See Note, page 1.

METRIC DOWEL PINS

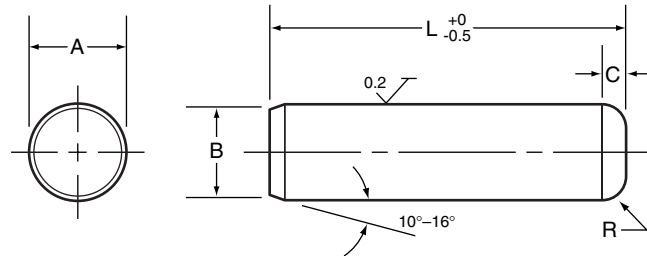
Hardened and Ground ■ Dimensions

Applicable or Similar Specifications: ASME B18.8.5M, ISO 8734 or DIN 6325.

Installation warning: Dowel pins should not be installed by striking or hammering. Wear safety glasses or shield when pressing chamfered point end first.

NOTES

- Material:** ASME B18.8.5M-alloy steel
- Hardness:** Rockwell C60 minimum (surface)
Rockwell C 50-58 (core)
- Shear Stress:** Calculated values based on 1050 MPa.
- Surface Finish:** 0.2 micrometer maximum



DIMENSIONS

APPLICATION DATA

nominal size	A pin diameter		B point diameter		C crown height max.	R crown radius min.	calculated single shear strength		recommended hole size	
	max.	min.	max.	min.			kN	pounds	max.	min.
3	3.008	3.003	2.9	2.6	0.8	0.3	7.4	1,670	3.000	2.987
4	4.009	4.004	3.9	3.6	0.9	0.4	13.2	2,965	4.000	3.987
5	5.009	5.004	4.9	4.6	1.0	0.4	20.6	4,635	5.000	4.987
6	6.010	6.004	5.8	5.4	1.1	0.4	29.7	6,650	6.000	5.987
8	8.012	8.006	7.8	7.4	1.3	0.5	52.5	11,850	8.000	7.987
10	10.012	10.006	9.8	9.4	1.4	0.6	82.5	18,550	10.000	9.987
12	12.013	12.007	11.8	11.4	1.6	0.6	119.0	26,700	12.000	11.985
16	16.013	16.007	15.8	15.3	1.8	0.8	211.0	47,450	16.000	15.985
20	20.014	20.008	19.8	19.3	2.0	0.8	330.0	74,000	20.000	19.983
25	25.014	25.008	24.8	24.3	2.3	1.0	515.0	116,000	25.000	24.983

All dimensions in millimeters.

METRIC SOCKET SET SCREWS ■ Knurled Cup Point and Plain Cup Point ■ Dimensions

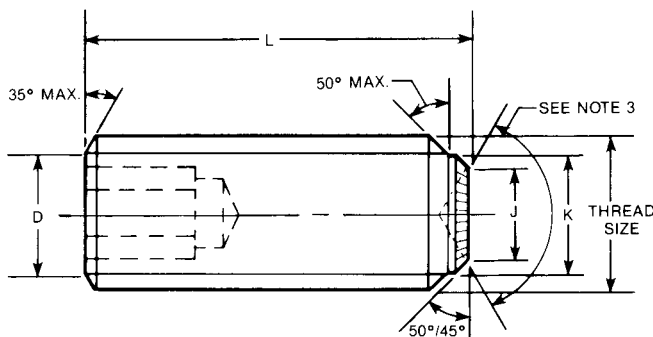
Threads: ANSI B 1.13M, ISO 261, ISO 262
(coarse series only)

Grade: 45H

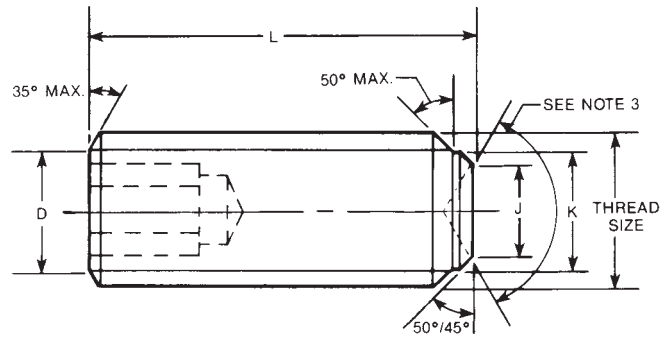
Applicable or Similar Specifications: ANSI B 18.3.6M,
ISO 4029, DIN 916, DIN 915, DIN 914, DIN 913

NOTES

- 1. Material:** ASTM F912M
- 2. Hardness:** Rockwell C45-53
- 3. Angle:** The cup angle is 135 maximum for screw lengths equal to or smaller than screw diameter. For longer lengths, the cup angle will be 124 maximum
- 4. Thread Class:** 4g 6g



KNURLED CUP POINT



PLAIN CUP POINT

LENGTH TOLERANCE

nominal screw length	nominal screw diameter
	M1.6 thru M24
	tolerance on lgth., mm
Up to 12 mm, incl.	±0.3
Over 12 to 50 mm, incl.	±0.5
Over 50 mm	±0.8

DIMENSIONS

APPLICATION DATA

nom. thread size	pitch	D max.	J max.		K max.	L min. preferred		W nom.	recommended* seating torque	
			plain cup	knurled cup		plain cup	knurled cup		N-m	in-lbs.
MICROSIZE – Plain Cup Only										
M1.6	0.35	1.0	0.80	–	–	2.0	–	0.7	0.09	0.8
M2	0.40	1.32	1.00	–	–	2.5	–	0.9	0.21	1.8
M2.5	0.45	1.75	1.25	–	–	3.0	–	1.3	0.57	5.0
STANDARD SIZE – Knurled Cup Point Supplied Unless Plain Cup Point Is Specified										
M3	0.5	2.10	1.50	1.40	2.06	3.0	3.0	1.5	0.92	8.0
M4	0.7	2.75	2.00	2.10	2.74	3.0	3.0	2.0	2.2	19.0
M5	0.8	3.70	2.50	2.50	3.48	4.0	4.0	2.5	4.0	35.0
M6	1.0	4.35	3.00	3.30	4.14	4.0	5.0	3.0	7.2	64
M8	1.25	6.00	5.00	5.00	5.62	5.0	6.0	4.0	17.0	150.0
M10	1.5	7.40	6.00	6.00	7.12	6.0	8.0	5.0	33.0	290
M12	1.75	8.60	8.00	8.00	8.58	8.0	10.0	6.0	54.0	480
M16	2.0	12.35	10.00	10.00	11.86	12.0	14.0	8.0	134	1190
M20	2.5	16.00	14.00	14.00	14.83	16.0	18.0	10.0	237	2100
M24	3.0	18.95	16.00	16.00	17.80	20.0	20.0	12.0	440	3860

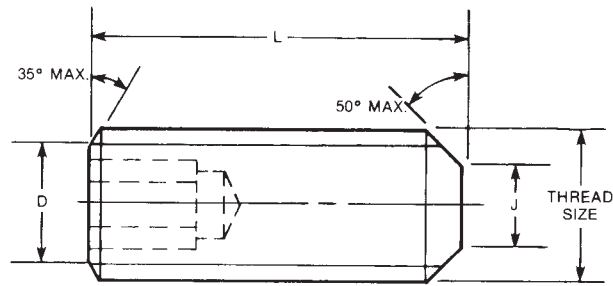
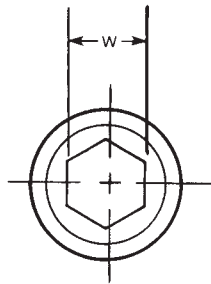
All dimensions in millimeters.

*Not applicable to screws with a length equal to or less than the diameter. See Note, page 1.

METRIC SOCKET SET SCREW

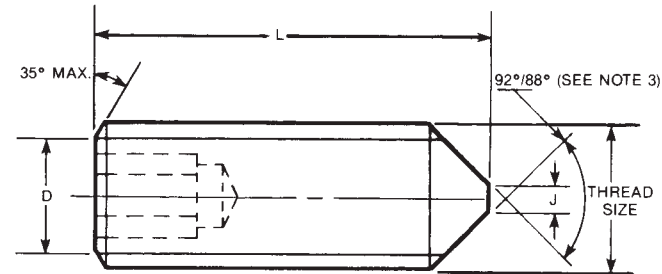
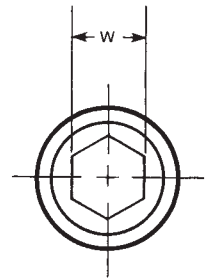
Flat Point, Cone Point, Dog Point Styles ■ Dimensions

REF. ISO 4026



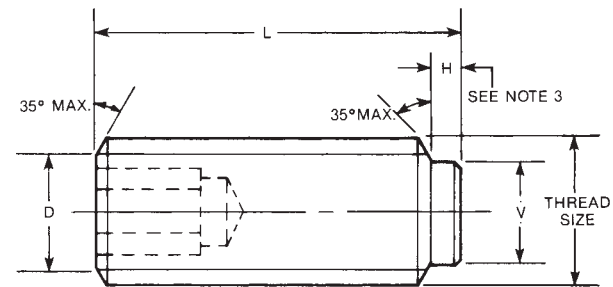
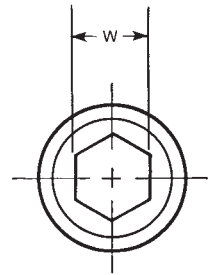
FLAT POINT

REF. ISO 4027



CONE POINT

REF. ISO 4028
ISO 7435



DOG POINT

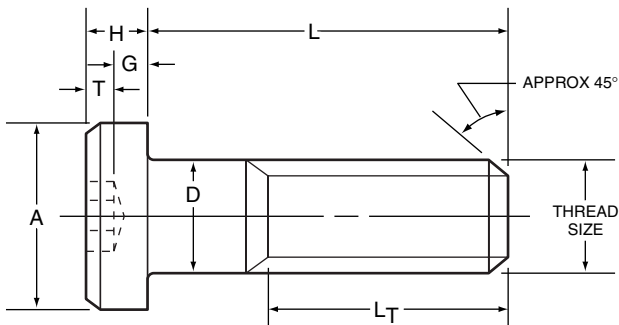
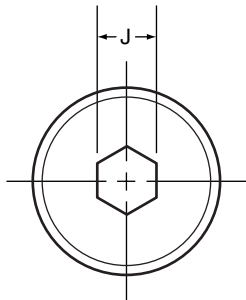
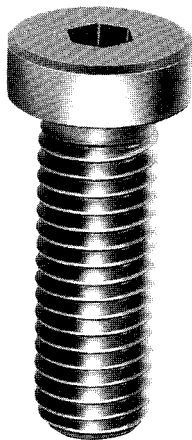
DIMENSIONS

nom. thread size	pitch	D max.	flat point		cone point		dog point			
			J	L	J	L	H		L min. preferred	V max.
			max.	min. preferred	max.	min. preferred	short lgth.	long lgth.		
M3	0.5	2.10	2.0	3.0	0.3	4.0	0.75	1.5	5.0	2.00
M4	0.7	2.75	2.5	3.0	0.4	4.0	1.00	2.0	5.0	2.50
M5	0.8	3.70	3.5	4.0	0.5	5.0	1.25	2.5	6.0	3.50
M6	1.00	4.25	4.0	4.0	1.5	6.0	1.50	3.0	6.0	4.00
M8	1.25	6.00	5.5	5.0	2.0	6.0	2.00	4.0	8.0	5.50
M10	1.50	7.40	7.0	6.0	2.5	8.0	2.50	5.0	8.0	7.00
M12	1.75	8.60	8.5	8.0	3.0	10.0	3.00	6.0	12.0	8.50
M16	2.00	12.35	12.0	12.0	4.0	14.0	4.00	8.0	16.0	12.00
M20	2.50	16.00	15.0	14.0	6.0	18.0	5.00	10.0	20.0	15.00
M24	3.00	18.95	18.0	20.0	8.0	20.0	6.00	12.0	22.0	18.00

METRIC LOW HEAD CAP SCREWS

Threads: ANSI B 1.13M, ISO 262
(coarse series only)
Property Class: 10.9
Similar Specifications: DIN 7984,
DIN 6912

- NOTES**
- 1. **Material:** ASTM A574M-alloy steel
 - 2. **Hardness:** Rc 33-39
 - 3. **Tensile Stress:** 1040 MPa
 - 4. **Yield Stress:** 940 MPa
 - 5. **Thread Class:** 4g 6g



DIMENSIONS

APPLICATION DATA

nom. thread size	pitch	A max.	D max.	G min.	T min.	H max.	L _T min.	J nom.	recommended* seating torque	
									plain	
									N-m	in.-lbs.
M4	0.7	7	4	1.06	1.48	2.8	20	3	4.5	40
M5	0.8	8.5	5	1.39	1.85	3.5	22	4	8.5	75
M6	1.0	10	6	1.65	2.09	4.0	24	5	14.5	130
M8	1.25	13	8	2.24	2.48	5.0	28	6	35	310
M10	1.5	16	10	2.86	3.36	6.5	32	8	70	620
M12	1.75	18	12	3.46	4.26	8.0	36	10	120	1060
M16	2.0	24	16	4.91	4.76	10.0	44	12	300	2650
M20	2.5	30	20	6.10	6.07	12.5	52	14	575	5100

All dimensions in millimeters.
*Torque calculated to induce 620 MPa in the screw threads.
Torque values are for plain screws. (See Note, page 1.)

METRIC HEXAGON KEYS

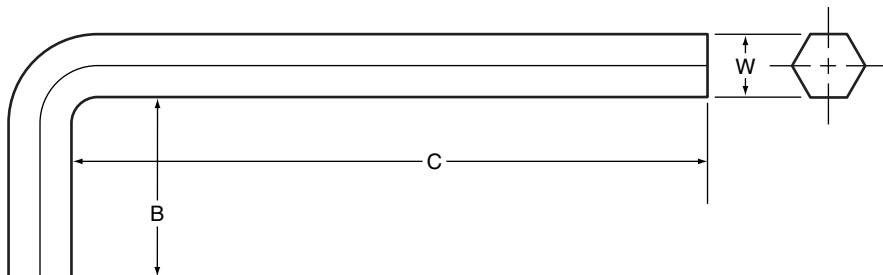
Dimensions ■ Mechanical Properties ■ Socket Applications

These UNBRAKO keys are made to higher requirements than ISO or DIN keys, which may not properly torque Class 12.9 cap screws. The strength and dimensional requirements are necessary to properly install the products in this catalog.

Material: ANSI B18.3.2.M alloy steel

Dimensions: ANSI B18.3.2.M

Similar Specifications: DIN 911, ISO 2936



METRIC KEY APPLICATION CHART

size W	socket cap screws		socket cap screws	flat head socket cap screws	button head shoulder screws	socket set screws
	std. head height	low head				
0.7 0.9 1.3						M1.6 M2 M2.5
1.5 2.0 2.5	M1.6/M2 M2.5 M3		M3 M4	M3 M4		M3 M4 M5
3.0 4.0 5.0	M4 M5 M6	M4 M5 M6	M5 M6 M8	M5 M6 M8	M6 M8 M10	M6 M8 M10
6.0 8.0 10.0	M8 M10 M12	M8 M10 M12	M10 M12 M16	M10 M12 M16	M12 M16 M20	M12 M16 M20
12.0 14.0 17.0	M14 M16 M20	M16 M20 M24	M20 M24		M24	M24
19.0 22.0 27.0	M24 M30 M36					
32.0 36.0	M42 M48					

DIMENSIONS

MECHANICAL PROPERTIES

key size W		B nominal	C nominal		torsional shear strength minimum		torsional yield strength minimum	
max.	min.		short arm	long arm	N-m	ln-lbs.	N-m	ln-lbs.
0.711 0.889 1.270	0.698 0.876 1.244	5.5 9 13.5	31 31 42	*69 71 75	0.12 0.26 0.73	1.1 2.3 6.5	0.1 0.23 .63	0.9 2. 5.6
1.500 2.000 2.500	1.470 1.970 2.470	14 16 18	45 50 56	78 83 90	1.19 2.9 5.4	10.5 26 48	1.02 2.4 4.4	9. 21 39
3.000 4.000 5.000	2.960 3.960 4.960	20 25 28	63 70 80	100 106 118	9.3 22.2 42.7	82 196 378	8. 18.8 36.8	71 166 326
6.000 8.000 10.000	5.960 7.950 9.950	32 36 40	90 100 112	140 160 170	74 183 345	655 1,620 3,050	64 158 296	566 1,400 2,620
12.000 14.000 17.000	11.950 13.930 16.930	45 55 60	125 140 160	212 236 250	634 945 1,690	5,610 8,360 15,000	546 813 1,450	4,830 7,200 12,800
19.000 22.000 24.000	18.930 21.930 23.930	70 80 90	180 *200 *224	280 *335 *375	2,360 3,670 4,140	20,900 32,500 36,600	2,030 3,160 3,560	18,000 28,000 31,500
27.000 32.000 36.000	26.820 31.820 35.820	100 125 140	*250 *315 *355	*500 *630 *710	5,870 8,320 11,800	51,900 73,600 104,000	5,050 7,150 10,200	44,700 63,300 90,300

All dimensions in millimeters.

*Non-stock sizes

ISO TOLERANCES FOR METRIC FASTENERS

nominal dimension		tolerance zone in mm (external measurements)													tolerance zone in mm					
over	to	h6	h8	h10	h11	h13	h14	h15	h16	js14	js15	js16	js17	m6	H7	H8	H9	H11	H13	H14
0	1	0 -0.006	0 -0.014	0 -0.040	0 -0.060	0 -0.14								+0.002 +0.008	+0.010 0	+0.0014 0	+0.025 0	+0.060 0	+0.14 0	
1	3	0 -0.006	0 -0.014	0 -0.040	0 -0.060	0 -0.14	0 -0.25	0 -0.40	0 -0.60	±0.125	±0.20	±0.30	±0.50	+0.002 +0.008	+0.010 0	+0.014 0	+0.025 0	+0.060 0	+0.14 0	+0.25 0
3	6	0 -0.008	0 -0.018	0 -0.048	0 -0.075	0 -0.18	0 -0.30	0 -0.48	0 -0.75	±0.15	±0.24	±0.375	±0.60	+0.004 +0.012	+0.012 0	+0.018 0	+0.030 0	+0.075 0	+0.18 0	+0.30 0
6	10	0 -0.009	0 -0.022	0 -0.058	0 -0.090	0 -0.22	0 -0.36	0 -0.58	0 -0.90	±0.18	±0.29	±0.45	±0.75	+0.006 +0.0015	+0.015 0	+0.022 0	+0.036 0	+0.090 0	+0.22 0	+0.36 0
10	18	0 -0.011	0 -0.027	0 -0.070	0 -0.110	0 -0.27	0 -0.43	0 -0.70	0 -1.10	±0.215	±0.35	±0.55	±0.90	+0.007 +0.018	+0.018 0	+0.027 0	+0.043 0	+0.110 0	+0.27 0	+0.43 0
18	30	0 -0.030	0 -0.033	0 -0.084	0 -0.130	0 -0.33	0 -0.52	0 -0.84	0 -1.30	±0.26	±0.42	±0.65	±1.05	+0.008 +0.021	+0.021 0	+0.033 0	+0.052 0	+0.130 0	+0.33 0	+0.52 0
30	50					0 -0.39	0 -0.62	0 -1.00	0 -1.60	±0.31	±0.50	±0.80	±1.25						+0.39 0	+0.62 0
50	80					0 -0.46	0 -0.74	0 -1.20	0 -1.90	±0.37	±0.60	±0.95	±1.50						+0.46 0	+0.74 0
80	120					0 -0.54	0 -0.87	0 -1.40	0 -2.20	±0.435	±0.70	±1.10	±1.75						+0.54 0	+0.87 0
120	180									±0.50	±0.80	±1.25	±2.00							
180	250									±0.575	±0.925	±1.45	±2.30							
250	315									±0.65	±1.05	±1.60	±2.60							
315	400									±0.70	±1.15	±1.80	±2.85							
400	500									±0.775	±1.25	±2.00	±3.15							

ISO TOLERANCES FOR SOCKET SCREWS

nominal dimension		tolerance zone in mm										
over	to	C13	C14	D9	D10	D11	D12	EF8	E11	E12	Js9	K9
	3	+0.20 +0.06	+0.31 +0.06	+0.045 +0.020	+0.060 +0.020	+0.080 +0.020	+0.12 +0.02	+0.024 +0.010	+0.074 +0.014	+0.100 +0.014	±0.0125	0 -0.025
3	6	+0.24 +0.06	+0.37 +0.07	+0.060 +0.030	+0.078 +0.030	+0.115 +0.030	+0.15 +0.03	+0.028 +0.014	+0.095 +0.020	+0.140 +0.020	±0.015	0 -0.030
6	10					+0.130 +0.040	+0.19 +0.40	+0.040 +0.018	+0.115 +0.025	+0.115 +0.025	±0.018	0 -0.036
10	18						+0.2 +0.05		+0.142 +0.032	+0.212 +0.032		
18	30						+0.275 +0.065					
30	50						+0.33 +0.08					
50	80						+0.40 +0.10					

References

ISO R 286
ISO 4759/I
ISO 4759/II
ISO 4759/III

Notes

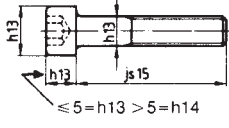
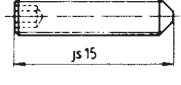
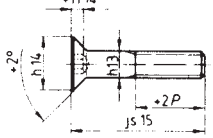
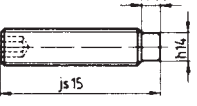
ANSI standards allow slightly wider tolerances for screw lengths than ISO and DIN.

The table is intended to assist in the design with metric fasteners. For tolerances not listed here refer to the complete standards.

Tolerances for Metric Fasteners

The tolerances in the tables below are derived from ISO standard: ISO 4759

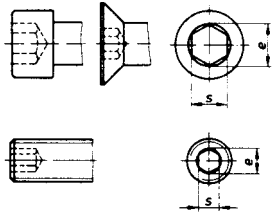
The tables show tolerances on the most common metric fasteners. However, occasionally some slight modifications are made.

Item	DIN	Item	DIN
	912		913 914 916
	7991		915 966

Notes

Product grade A applies to sizes up to M24 and length not exceeding 10 x diameter or 150 mm, whatever is shorter.

Product grade B applies to the sizes above M24 and all sizes with lengths, greater than 10 x diameter or 150 mm, whichever is shorter.

Feature	Tolerance	
Hexagon Sockets	s	* **
	0.7	EF8
	0.9	JS9
	1.3	K9
	1.5	D9
	2	
	2.5	D10
	3	D11
	4	E11
	5	
	6	E12
	8	
	10	
	12	
	14	
	>14	D12

*Tolerance zones for socket set screws

**Tolerance zones for socket head cap screws

Note: For S 0.7 to 1.3 the actual allowance in the product standards has been slightly modified for technical reasons.

CONVERSION CHART

SI UNITS & CONVERSIONS FOR CHARACTERISTICS OF MECHANICAL FASTENERS						
property	unit	symbol	conversion			approximate equivalent
			from	to	multiply by	
length	meter	m	inch	mm	25.4	25mm = 1 in.
	centimeter	cm	inch	cm	2.54	300mm = 1ft.
	millimeter	mm	foot	mm	304.8	1m = 39.37 in.
mass	kilogram	kg	once	g	28.35	28g = 1 oz.
	gram	g	pound	kg	.4536	1kg = 2.2 lb. = 35 oz.
	tonne (megagram)	t	ton (2000 lb)	kg	907.2	1t = 2200 lbs.
density	kilogram per cubic meter	kg/m ³	pounds per cu. ft.	kg/m	16.02	16kg/m = 1 lb/ft. ³
temperature	deg. Celsius	°C	deg. Fahr.	°C	(°F – 32) x 5/9	0°C = 32°F 100°C = 212°F
area	square meter	m ²	sq. in.	mm ²	645.2	645mm ² = 1 in. ²
	square millimeter	mm ²	sq. ft.	m ²	.0929	1m ² = 11 ft. ²
volume	cubic meter	m ³	cu. in.	mm ³	16387	16400mm ³ = 1 in. ³
	cubic centimeter	cm ³	cu.ft.	m ³	.02832	1m ³ = 35 ft. ³
	cubic millimeter	mm ³	cu. yd.	m ³	.7645	1m ³ = 1.3 yd. ³
force	newton	N	ounce (Force)	N	.278	1N = 3.6 ozf
	kilonewton	kN	pound (Force)	kN	.00445	4.4N = 1 lbf
	meganewton	MN	Kip	MN	.00445	1kN = 225 lbf
stress	megapascal	MPa	pound/in ² (psi)	MPa	.0069	1MPa = 145 psi
	newtons/sq.m	N/m ²	Kip/in ² (ksi)	MPa	6.895	7MPa = 1 ksi
torque	newton-meter	N•m	inch-ounce	N-m	.00706	1N•m = 140 in. oz.
			inch-pound	N-m	.113	1N•m = 9 in. lb.
			foot-pound	N-m	1.356	1N•m = .75 ft. lb. 1.4 N•m = 1 ft. lb.

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IMPORTANT

The technical discussions represent typical applications only. The use of the information is at the sole discretion of the reader. Because applications vary enormously, UNBRAKO does not warrant the scenarios described are appropriate for any specific application. The reader must consider all variables prior to using this information.

INSTALLATION CONTROL

Several factors should be considered in designing a joint or selecting a fastener for a particular application.

JOINT DESIGN AND FASTENER SELECTION.

Joint Length

The longer the joint length, the greater the total elongation will occur in the bolt to produce the desired clamp load or preload. In design, if the joint length is increased, the potential loss of preload is decreased.

Joint Material

If the joint material is relatively stiff compared to the bolt material, it will compress less and therefore provide a less sensitive joint, less sensitive to loss of preload as a result of brinelling, relaxation and even loosening.

Thread Stripping Strength

Considering the material in which the threads will be tapped or the nut used, there must be sufficient engagement length to carry the load. Ideally, the length of thread engagement should be sufficient to break the fastener in tension. When a nut is used, the wall thickness of the nut as well as its length must be considered.

An estimate, a calculation or joint evaluation will be required to determine the tension loads to which the bolt and joint will be exposed. The size bolt and the number necessary to carry the load expected, along with the safety factor, must also be selected.

The safety factor selected will have to take into consideration the consequence of failure as well as the additional holes and fasteners. Safety factors, therefore, have to be determined by the designer.

SHEAR APPLICATIONS

Shear Strength of Material

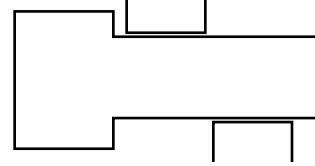
Not all applications apply a tensile load to the fastener. In many cases, the load is perpendicular to the fastener in shear. Shear loading may be single, double or multiple loading.

There is a relationship between the tensile strength of a material and its shear strength. For alloy steel, the shear strength is 60% of its tensile strength. Corrosion resistant steels (e.g. 300-Series stainless steels) have a lower tensile/shear relationship and it is usually 50-55%

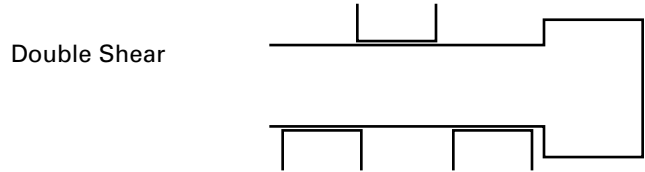
Single/Double Shear

Single shear strength is exactly one-half the double shear value. Shear strength listed in pounds per square inch (psi) is the shear load in pounds multiplied by the cross sectional area in square inches.

Single Shear



Double Shear



OTHER DESIGN CONSIDERATIONS

Application Temperature

For elevated temperature, standard alloy steels are useful to about 550°F–600°F. However, if plating is used, the maximum temperature may be less (eg. cadmium should not be used over 450°F).

Austenitic stainless steels (300 Series) may be useful to 800°F. They can maintain strength above 800°F but will begin to oxidize on the surface.

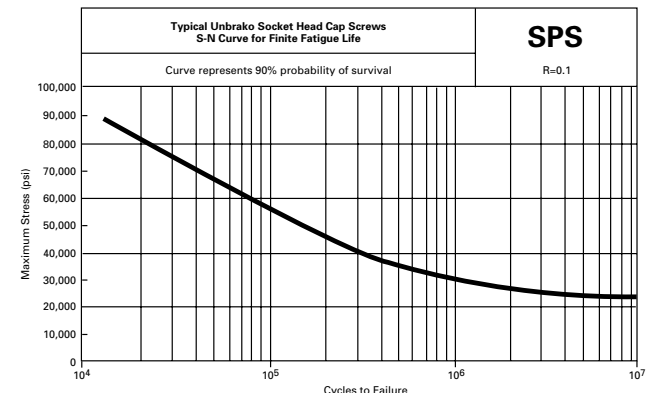
Corrosion Environment

A plating may be selected for mild atmospheres or salts. If plating is unsatisfactory, a corrosion resistant fastener may be specified. The proper selection will be based upon the severity of the corrosive environment.

FATIGUE STRENGTH

S/N Curve

Most comparative fatigue testing and specification fatigue test requirements are plotted on an S/N curve. In this curve, the test stress is shown on the ordinate (y-axis) and the number of cycles is shown on the abscissa (x-axis) in a logarithmic scale. On this type curve, the high load to low load ratio must be shown. This is usually $R = .1$, which means the low load in all tests will be 10% of the high load.



Effect of Preload

Increasing the R to .2, .3 or higher will change the curve shape. At some point in this curve, the number of cycles will reach 10 million cycles. This is considered the

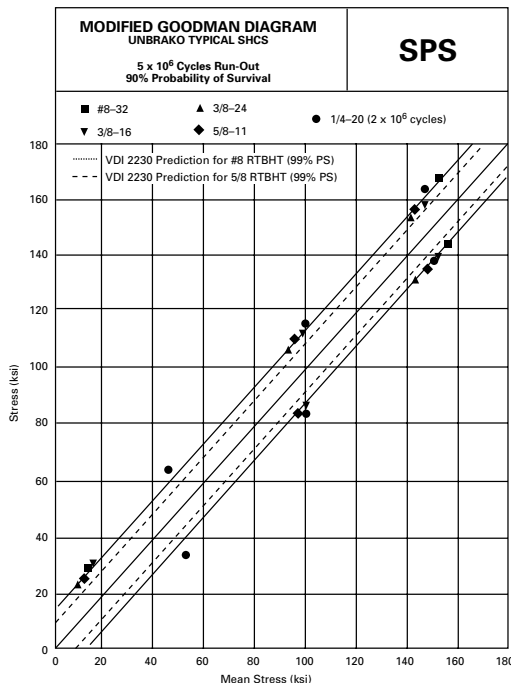
SCREW FASTENER THEORY & APPLICATIONS

endurance limit or the stress at which infinite life might be expected.

Modified Goodman/ Haigh Soderberg Curve

The S/N curve and the information it supplies will not provide the information needed to determine how an individual fastener will perform in an actual application. In application, the preload should be higher than any of the preloads on the S/N curve.

Therefore, for application information, the modified Goodman Diagram and/or the Haigh Soderberg Curve are more useful. These curves will show what fatigue performance can be expected when the parts are properly preloaded.



METHODS OF PRELOADING

Elongation

The modulus for steel of 30,000,000 (thirty million) psi means that a fastener will elongate .001 in/in of length for every 30,000 psi in applied stress. Therefore, if 90,000 psi is the desired preload, the bolt must be stretched .003 inches for every inch of length in the joint.

This method of preloading is very accurate but it requires that the ends of the bolts be properly prepared and also that all measurements be very carefully made. In addition, direct measurements are only possible where both ends of the fastener are available for measurement after installation. Other methods of measuring length changes are ultrasonic, strain gages and turn of the nut.

Torque

By far, the most popular method of preloading is by torque. Fastener manufacturers usually have recommended seating torques for each size and material fastener. The only requirement is the proper size torque wrench, a conscientious operator and the proper torque requirement.

Strain

Since stress/strain is a constant relationship for any given material, we can use that relationship just as the elongation change measurements were used previously.

Now, however, the strain can be detected from strain gages applied directly to the outside surface of the bolt or by having a hole drilled in the center of the bolt and the strain gage installed internally. The output from these gages need instrumentation to convert the gage electrical measurement method. It is, however, an expensive method and not always practical.

Turn of the Nut

The nut turn method also utilizes change in bolt length. In theory, one bolt revolution (360° rotation) should increase the bolt length by the thread pitch. There are at least two variables, however, which influence this relationship. First, until a snug joint is obtained, no bolt elongation can be measured. The snugging produces a large variation in preload. Second, joint compression is also taking place so the relative stiffnesses of the joint and bolt influences the load obtained.

VARIABLES IN TORQUE

Coefficient of Friction

Since the torque applied to a fastener must overcome all friction before any loading takes place, the amount of friction present is important.

In a standard unlubricated assembly, the friction to be overcome is the head bearing area and the thread-to-thread friction. Approximately 50% of the torque applied will be used to overcome this head-bearing friction and approximately 35% to overcome the thread friction. So 85% of the torque is overcoming friction and only 15% is available to produce bolt load.

If these interfaces are lubricated (cadmium plate, molybdenum disulfide, anti-seize compounds, etc.), the friction is reduced and thus greater preload is produced with the same torque.

The change in the coefficient of friction for different conditions can have a very significant effect on the slope of the torque tension curve. If this is not taken into consideration, the proper torque specified for a plain unlubricated bolt may be sufficient to yield or break a lubricated fastener.

Thread Pitch

The thread pitch must be considered when a given stress is to be applied, since the cross-sectional area used for stress calculations is the thread tensile stress area and is different for coarse and fine threads. The torque recommendations, therefore, are slightly higher for fine threads than for coarse threads to achieve the same stress.

Differences between coarse and fine threads.

Coarse Threads are...

- more readily available in industrial fasteners.
- easier to assemble because of larger helix angle.
- require fewer turns and reduce cross threading.
- higher thread stripping strength per given length.
- less critical of tap drill size.
- not as easily damaged in handling.

Their disadvantages are...

- lower tensile strength.
- reduced vibrational resistance.
- coarse adjustment.

Fine Threads provide...

- higher tensile strength.
- greater vibrational resistance.
- finer adjustment.

Their disadvantages are...

- easier cross threaded.
- threads damaged more easily by handling.
- tap drill size slightly more critical.
- slightly lower thread stripping strength.

Other Design Guidelines

In addition to the joint design factors discussed, the following considerations are important to the proper use of high-strength fasteners.

- Adequate thread engagement should be guaranteed by use of the proper mating nut height for the system. Minimum length of engagement recommended in a tapped hole depends on the strength of the material, but in all cases should be adequate to prevent stripping.
- Specify nut of proper strength level. The bolt and nut should be selected as a system.
- Specify compatible mating female threads. 2B tapped holes or 3B nuts are possibilities.
- Corrosion, in general, is a problem of the joint, and not just of the bolt alone. This can be a matter of galvanic action between dissimilar metals. Corrosion of the fastener material surrounding the bolt head or nut can be critical with high-strength bolting. Care must be exercised in the compatibility of joint materials and/or coatings to protect dissimilar metals.

PROCESSING CONTROL

The quality of the raw material and the processing control will largely affect the mechanical properties of the finished parts.

MATERIAL SELECTION

The selection of the type of material will depend on its end use. However, the control of the analysis and quality is a critical factor in fastener performance. The material must yield reliable parts with few hidden defects such as cracks, seams, decarburization and internal flaws.

FABRICATION METHOD

Head

There are two general methods of making bolt heads, forging and machining. The economy and grain flow resulting from forging make it the preferred method.

The temperature of forging can vary from room temperature to 2000°F. By far, the greatest number of parts are cold upset on forging machines known as headers or boltmakers. For materials that do not have enough formability for cold forging, hot forging is used. Hot forging is also used for bolts too large for cold upsetting due to machine capacity. The largest cold forging

machines can make bolts up to 1-1/2 inch diameter. For large quantities of bolts, hot forging is more expensive than cold forging.

Some materials, such as stainless steel, are warm forged at temperatures up to 1000°F. The heating results in two benefits, lower forging pressures due to lower yield strength and reduced work hardening rates.

Machining is the oldest method and is used for very large diameters or small production runs.

The disadvantage is that machining cuts the metal grain flow, thus creating planes of weakness at the critical head-to-shank fillet area. This can reduce tension fatigue performance by providing fracture planes.

Fillet

The head-to-shank transition (fillet) represents a sizable change in cross section at a critical area of bolt performance. It is important that this notch effect be minimized. A generous radius in the fillet reduces the notch effect. However, a compromise is necessary because too large a radius will reduce load-bearing area under the head.

Composite radii such as elliptical fillets, maximize curvature on the shank side of the fillet and minimize it on the head side to reduce loss of bearing area on the load-bearing surface.

Critical Fastener Features

Head-Shank-Fillet: This area on the bolt must not be restricted or bound by the joint hole. A sufficient chamfer or radius on the edge of the hole will prevent interference that could seriously reduce fatigue life. Also, if the bolt should seat on an unchamfered edge, there might be serious loss of preload if the edge breaks under load.

Threads

Threads can be produced by grinding, cutting or rolling.

In a rolled thread, the material is caused to flow into the thread die contour, which is ground into the surface during the manufacture of the die. Machines with two or three circular dies or two flat dies are most common.

Thread cutting requires the least tooling costs and is by far the most popular for producing internal threads. It is the most practical method for producing thin wall parts and the only technique available for producing large diameter parts (over 3 inches in diameter).

Thread grinding yields high dimensional precision and affords good control of form and finish. It is the only practical method for producing thread plug gages.

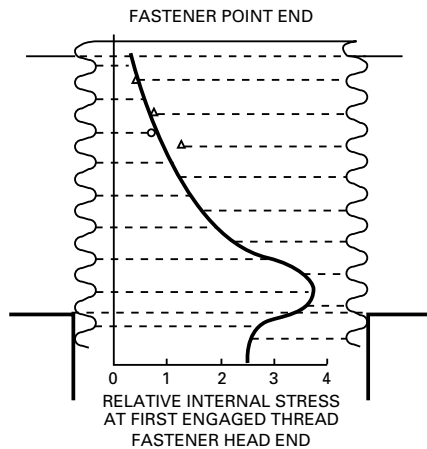
Both machining and grinding have the disadvantage of cutting material fibers at the most critical point of performance.

The shape or contour of the thread has a great effect on the resulting fatigue life. The thread root should be large and well rounded without sharp corners or stress risers. Threads with larger roots should always be used for harder materials.

In addition to the benefits of grain flow and controlled shape in thread rolling, added fatigue life can result when the rolling is performed after heat treatment.

SCREW FASTENER THEORY & APPLICATIONS

This is the accepted practice for high fatigue performance bolts such as those used in aircraft and space applications.



EVALUATING PERFORMANCE

Mechanical Testing

In the fastener industry, a system of tests and examinations has evolved which yields reliable parts with proven performance.

Some tests are conducted on the raw material; some on the finished product.

There always seems to be some confusion regarding mechanical versus metallurgical properties. Mechanical properties are those associated with elastic or inelastic reaction when force is applied, or that involve the relationship between stress and strain. Tensile testing stresses the fastener in the axial direction. The force at which the fastener breaks is called the breaking load or ultimate tensile strength. Load is designated in pounds, stress in pounds per square inch and strain in inches per inch.

When a smooth tensile specimen is tested, the chart obtained is called a Stress-Strain Curve. From this curve, we can obtain other useful data such as yield strength. The method of determining yield is known as the offset method and consists of drawing a straight line parallel to the stress strain curve but offset from the zero point by a specified amount. This value is usually 0.2% on the strain ordinate. The yield point is the intersection of the stress-strain curve and the straight line. This method is not applicable to fasteners because of the variables introduced by their geometry.

When a fastener tensile test is plotted, a load/elongation curve can be obtained. From this curve, a yield determination known as Johnson's 2/3 approximate method for determination of yield strength is used to establish fastener yield, which will be acceptable for design purposes. It is not recommended for quality control or specification requirements.

Torque-tension testing is conducted to correlate the required torque necessary to induce a given load in a mechanically fastened joint. It can be performed by hand or machine. The load may be measured by a tensile machine, a load cell, a hydraulic tensile indicator or by a strain gage.

Fatigue tests on threaded fasteners are usually alternating tension-tension loading. Most testing is done at more severe strain than its designed service load but usually below the material yield strength.

Shear testing, as previously mentioned, consists of loading a fastener perpendicular to its axis. All shear testing should be accomplished on the unthreaded portion of the fastener.

Checking hardness of parts is an indirect method for testing tensile strength. Over the years, a correlation of tensile strength to hardness has been obtained for most materials. See page 83 for more detailed information. Since hardness is a relatively easy and inexpensive test, it makes a good inspection check. In hardness checking, it is very important that the specimen be properly prepared and the proper test applied.

Stress durability is used to test parts which have been subjected to any processing which may have an embrittling effect. It requires loading the parts to a value higher than the expected service load and maintaining that load for a specified time after which the load is removed and the fastener examined for the presence of cracks.

Impact testing has been useful in determining the ductile brittle transformation point for many materials. However, because the impact loading direction is transverse to a fastener's normal longitude loading, its usefulness for fastener testing is minimal. It has been shown that many fastener tension impact strengths do not follow the same pattern or relationship of Charpy or Izod impact strength.

Metallurgical Testing

Metallurgical testing includes chemical composition, microstructure, grain size, carburization and decarburization, and heat treat response.

The chemical composition is established when the material is melted. Nothing subsequent to that process will influence the basic composition.

The microstructure and grain size can be influenced by heat treatment. Carburization is the addition of carbon to the surface which increases hardness. It can occur if heat treat furnace atmospheres are not adequately controlled. Decarburization is the loss of carbon from the surface, making it softer. Partial decarburization is preferable to carburization, and most industrial standards allow it within limits.

In summary, in order to prevent service failures, many things must be considered:

The Application Requirements

Strength Needed – Safety Factors

- Tension/Shear/Fatigue
- Temperature
- Corrosion
- Proper Preload

The Fastener Requirements

- Material
- Fabrication Controls
- Performance Evaluations

AN EXPLANATION OF JOINT DIAGRAMMS

When bolted joints are subjected to external tensile loads, what forces and elastic deformation really exist? The majority of engineers in both the fastener manufacturing and user industries still are uncertain. Several papers, articles, and books, reflecting various stages of research into the problem have been published and the volume of this material is one reason for confusion. The purpose of this article is to clarify the various explanations that have been offered and to state the fundamental concepts which apply to forces and elastic deformations in concentrically loaded joints. The article concludes with general design formulae that take into account variations in tightening, preload loss during service, and the relation between preloads, external loads and bolt loads.

The Joint Diagram

Forces less than proof load cause elastic strains. Conversely, changes in elastic strains produce force variations. For bolted joints this concept is usually demonstrated by joint diagrams.

The most important deformations within a joint are elastic bolt elongation and elastic joint compression in the axial direction. If the bolted joint in Fig. 1 is subjected to the preload F_i the bolt elongates as shown by the line OB in Fig. 2A and the joint compresses as shown by the line OJ. These two lines, representing the spring characteristics of the bolt and joint, are combined into one diagram in Fig. 2B to show total elastic deformation.

If a concentric external load F_e is applied under the bolt head and nut in Fig. 1, the bolt elongates an additional amount while the compressed joint members partially relax. These changes in deformation with external loading are the key to the interaction of forces in bolted joints.

In Fig. 3A the external load F_e is added to the joint diagram F_e is located on the diagram by applying the upper end to an extension of OB and moving it in until the lower end contacts OJ. Since the total amount of elastic deformation (bolt plus joint) remains constant for a given preload, the external load changes the total bolt elongation to $\Delta l_B + \lambda$ and the total joint compression to $\Delta l_J - \lambda$.

In Fig. 3B the external load F_e is divided into an additional bolt load F_{eB} and the joint load F_{eJ} , which unloads the compressed joint members. The maximum bolt load is the sum of the load preload and the additional bolt load:

$$F_{B \max} = F_i + F_{eB}$$

If the external load F_e is an alternating load, F_{eB} is that part of F_e working as an alternating bolt load, as shown in Fig. 3B. This joint diagram also illustrates that the joint absorbs more of the external load than the bolt subjected to an alternating external load.

The importance of adequate preload is shown in Fig. 3C. Comparing Fig. 3B and Fig. 3C, it can be seen that F_{eB} will remain relatively small as long as the preload F_i is greater than F_{eJ} . Fig. 3C represents a joint with insufficient preload. Under this condition, the amount of external load that the joint can absorb is limited, and the excess

load must then be applied to the bolt. If the external load is alternating, the increased stress levels on the bolt produce a greatly shortened fatigue life.

When seating requires a certain minimum force or when transverse loads are to be transformed by friction, the minimum clamping load $F_{J \min}$ is important.

$$F_{J \min} = F_{B \max} - F_e$$

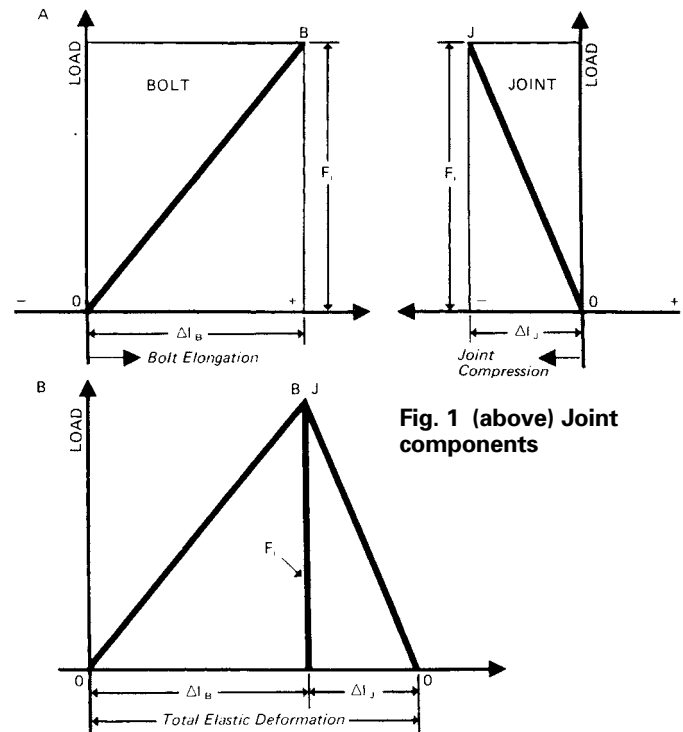


Fig. 1 (above) Joint components

Fig. 2 Joint diagram is obtained by combining load vs. deformation diagrams of bolt and joints.

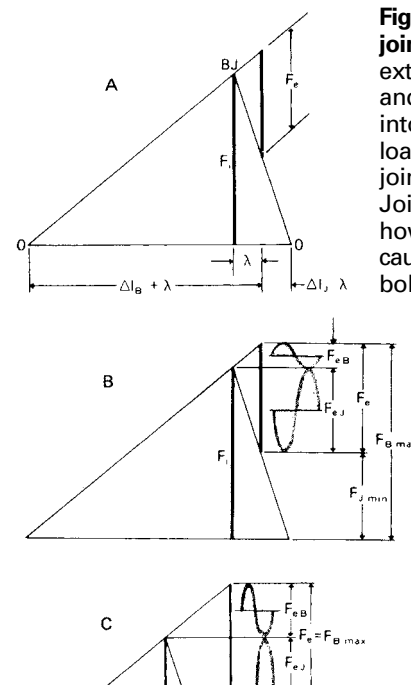


Fig. 3 The complete simple joint diagrams show external load F_e added (A), and external load divided into an additional bolt load F_{eB} and reduction in joint compression F_{eJ} . Joint diagram (C) shows how insufficient preload F_i causes excessive additional bolt load F_{eB} .

Spring Constants

To construct a joint diagram, it is necessary to determine the spring rates of both bolt and joint. In general, spring rate is defined as:

$$K = \frac{F}{\Delta l}$$

From Hook's law:

$$\Delta l = \frac{F l}{E A}$$

Therefore:

$$K = \frac{E A}{l}$$

To calculate the spring rate of bolts with different cross sections, the reciprocal spring rates, or compliances, of each section are added:

$$\frac{1}{K_B} = \frac{1}{K_1} + \frac{1}{K_2} + \dots + \frac{1}{K_n}$$

Thus, for the bolt shown in Fig. 4:

$$\frac{1}{K_B} = \frac{1}{E} \left(\frac{0.4d}{A_1} + \frac{l_1}{A_1} + \frac{l_2}{A_2} + \frac{l_3}{A_m} + \frac{0.4d}{A_m} \right)$$

where

d = the minor thread diameter and

A_m = the area of the minor thread diameter

This formula considers the elastic deformation of the head and the engaged thread with a length of $0.4d$ each.

Calculation of the spring rate of the compressed joint members is more difficult because it is not always obvious which parts of the joint are deformed and which are not. In general, the spring rate of a clamped part is:

$$K_J = \frac{E A_s}{l_j}$$

where A_s is the area of a substitute cylinder to be determined.

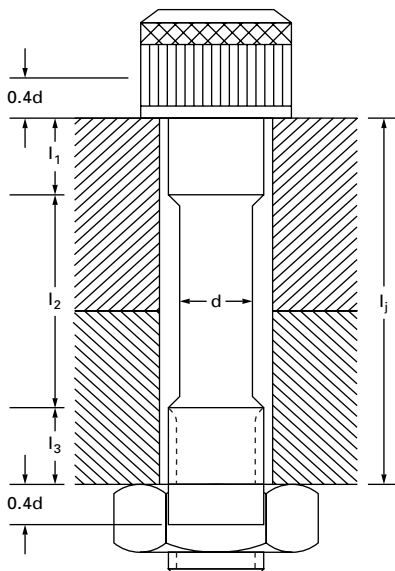


Fig. 4 Analysis of bolt lengths contributing to the bolt spring rate.

When the outside diameter of the joint is smaller than or equal to the bolt head diameter, i.e., as in a thin bushing, the normal cross sectioned area is computed:

$$A_s = \frac{\pi}{4} (D_c^2 - D_h^2)$$

where

D_c = OD of cylinder or bushing and

D_h = hole diameter

When the outside diameter of the joint is larger than head or washer diameter D_h , the stress distribution is in the shape of a barrel, Fig 5. A series of investigations proved that the areas of the following substitute cylinders are close approximations for calculating the spring contents of concentrically loaded joints.

When the joint diameter D_j is greater than D_h but less than $3D_h$:

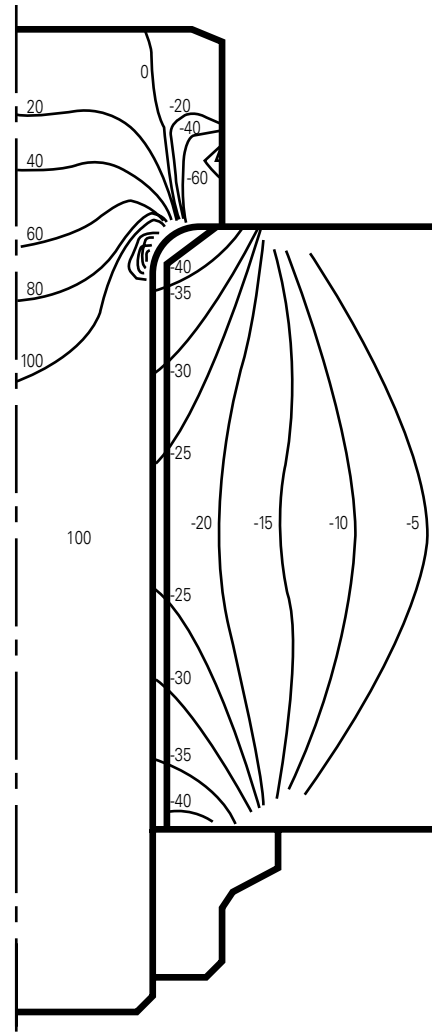


Fig. 5 Lines of equal axial stresses in a bolted joint obtained by the axisymmetric finite element method are shown for a 9/16-18 bolt preloaded to 100 KSI. Positive numbers are tensile stresses in KSI; negative numbers are compressive stresses in KSI.

$$A_s = \frac{\pi}{4} (D_H^2 - D_h^2) + \frac{\pi}{8} \left(\frac{D_J}{D_H} - 1 \right) \left(\frac{D_H l_J}{5} + \frac{l_J^2}{100} \right)$$

When the joint diameter D_J is equal to or greater than $3D_H$:

$$A_s = \frac{\pi}{4} [(D_H + 0.1 l_J)^2 - D_h^2]$$

These formulae have been verified in laboratories by finite element method and by experiments.

Fig. 6 shows joint diagrams for springy bolt and stiff joint and for a stiff bolt and springy joint. These diagrams demonstrate the desirability of designing with springy bolt and a stiff joint to obtain a low additional bolt load F_{eB} and thus a low alternating stress.

The Force Ratio

Due to the geometry of the joint diagram, Fig. 7,

$$F_{eB} = \frac{F_e K_B}{K_B + K_J}$$

$$\text{Defining } \Phi = \frac{K_B}{K_B + K_J}$$

$$F_{eB} = F_e \Phi \text{ and } \Phi, \text{ called the Force Ratio,} = \frac{F_{eB}}{F_e}$$

For complete derivation of Φ , see Fig. 7.

To assure adequate fatigue strength of the selected fastener the fatigue stress amplitude of the bolt resulting from an external load F_e is computed as follows:

$$\sigma_B = \pm \frac{F_{eB}/2}{A_m} \quad \text{or}$$

$$\sigma_B = \pm \frac{\Phi F_e}{2 A_m}$$

Effect of Loading Planes

The joint diagram in Fig 3, 6 and 7 is applicable only when the external load F_e is applied at the same loading planes as the preloaded F_i , under the bolt head and the nut. However, this is a rare case, because the external load usually affects the joint somewhere between the center of the joint and the head and the nut.

When a preloaded joint is subjected to an external load F_e at loading planes 2 and 3 in Fig. 8, F_e relieves the compression load of the joint parts between planes 2 and 3. The remainder of the system, the bolt and the joint parts between planes 1-2 and 3-4, feel additional load due to F_e applied planes 2 and 3, the joint material between planes 2 and 3 is the clamped part and all other joint members, fastener and remaining joint material, are clamping parts. Because of the location of the loading planes, the joint diagram changes from black line to the blue line. Consequently, both the additional bolt load $F_{B \max}$ decrease significantly when the loading planes of F_e shift from under the bolt head and nut toward the joint center.

Determination of the length of the clamped parts is, however, not that simple. First, it is assumed that the external load is applied at a plane perpendicular to the bolt axis. Second, the distance of the loading planes from each other has to be estimated. This distance may be expressed as the ratio of the length of clamped parts to the total joint length. Fig. 9 shows the effect of two different loading planes on the bolt load, both joints having the same preload F_i and the same external load F_e . The lengths of the clamped parts are estimated to be $0.75l_J$ for joint A, and $0.25l_J$ for joint B.

In general, the external bolt load is somewhere between $F_{eB} = 1\Phi F_e$ for loading planes under head and nut and $F_{eB} = 0\Phi F_e = 0$ when loading planes are in the joint center, as shown in Fig. 10. To consider the loading planes in calculations, the formula:

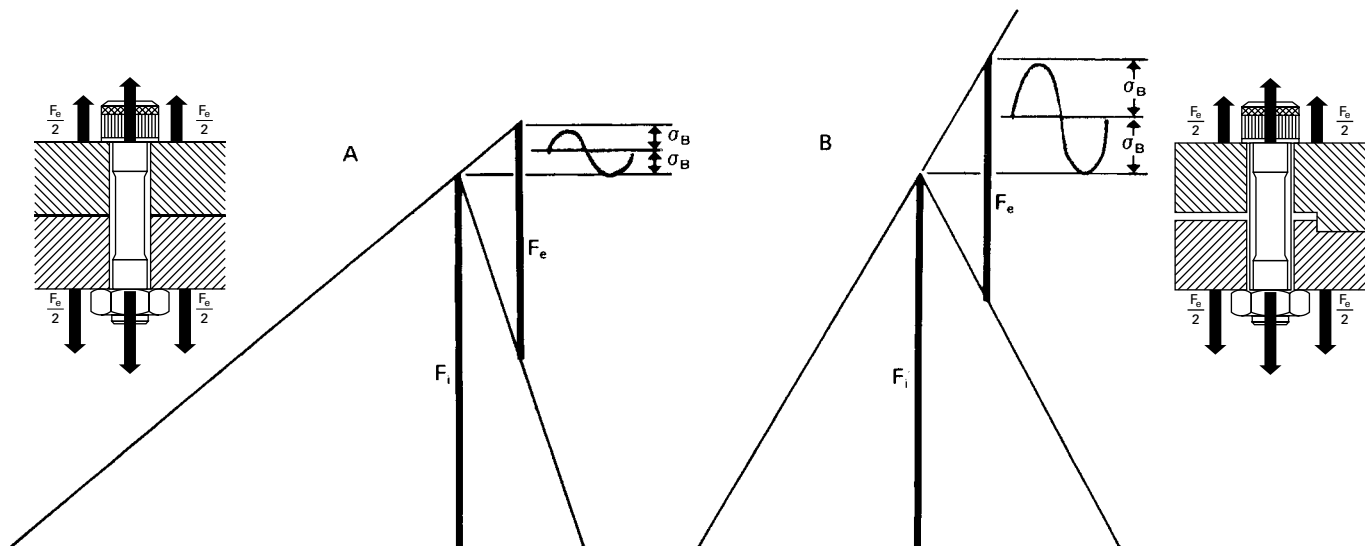


Fig. 6 Joint diagram of a springy bolt in a stiff joint (A), is compared to a diagram of a stiff bolt in a springy joint (B). Preload F_i and external load F_e are the same but diagrams show that alternating bolt stresses are significantly lower with a spring bolt in a stiff joint.

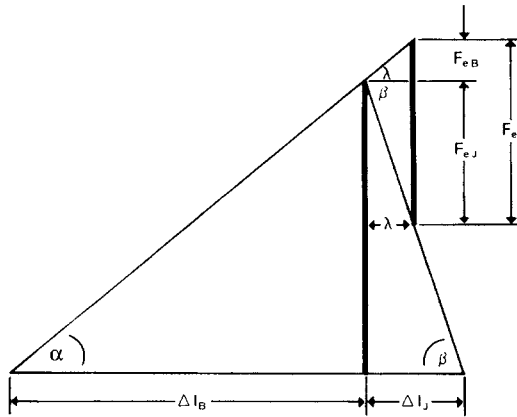


Fig. 7 Analysis of external load F_e and derivation of Force Ratio Φ .

$$\tan \alpha = \frac{F_i}{\Delta l_B} = K_B \text{ and } \tan \beta = \frac{F_i}{\Delta l_J} = K_J$$

$$\lambda = \frac{F_{eB}}{\tan \alpha} = \frac{F_{eJ}}{\tan \beta} = \frac{F_{eB}}{K_B} = \frac{F_{eJ}}{K_J} \text{ or}$$

$$F_{eJ} = \lambda \tan \beta \text{ and } F_{eB} = \lambda \tan \alpha$$

Since $F_e = F_{eB} + F_{eJ}$
 $F_e = F_{eB} + \lambda \tan \beta$

Substituting $\frac{F_{eB}}{\tan \alpha}$ for λ produces:

$$F_e = F_{eB} + \frac{F_{eB} \tan \beta}{\tan \alpha}$$

Multiplying both sides by $\tan \alpha$:

$$F_e \tan \alpha = F_{eB} (\tan \alpha + \tan \beta) \text{ and}$$

$$F_{eB} = \frac{F_e \tan \alpha}{\tan \alpha + \tan \beta}$$

Substituting K_B for $\tan \alpha$ and K_J for $\tan \beta$

$$F_{eB} = F_e \frac{K_B}{K_B + K_J}$$

$$\text{Defining } \Phi = \frac{K_B}{K_B + K_J}$$

$$F_{eB} = \Phi F_e$$

$$\Phi = \frac{F_{eB}}{F_e} \text{ and it becomes obvious why } \Phi \text{ is called force ratio.}$$

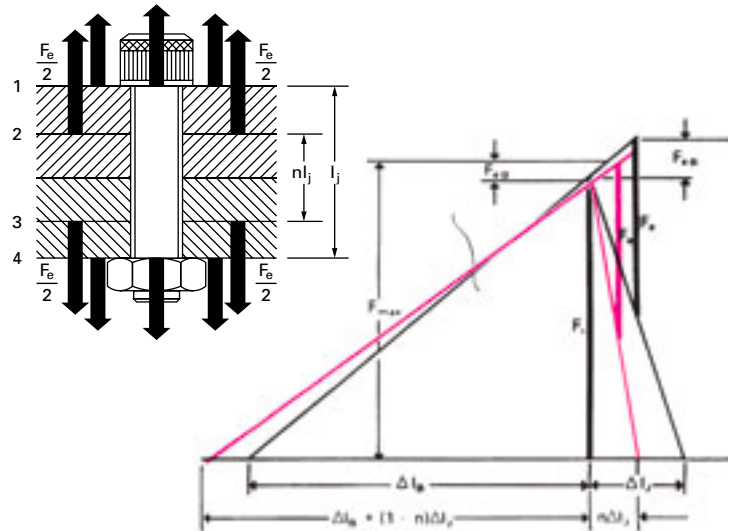


Fig. 8 Joint diagram shows effect of loading planes of F_e on bolt loads F_{eB} and $F_{B \max}$. Black diagram shows F_{eB} and $F_{B \max}$ resulting from F_e applied in planes 1 and 4. Orange diagram shows reduced bolt loads when F_e is applied in planes 2 and 3.

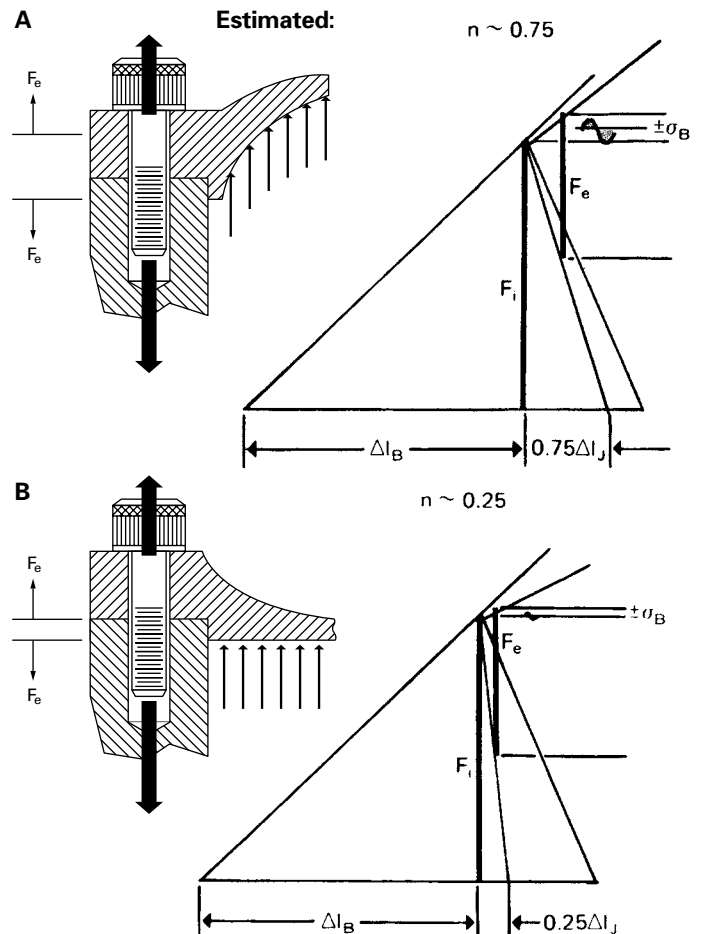


Fig. 9 When external load is applied relatively near bolt head, joint diagram shows resulting alternating stress α_B (A). When same value external load is applied relatively near joint center, lower alternating stress results (B).

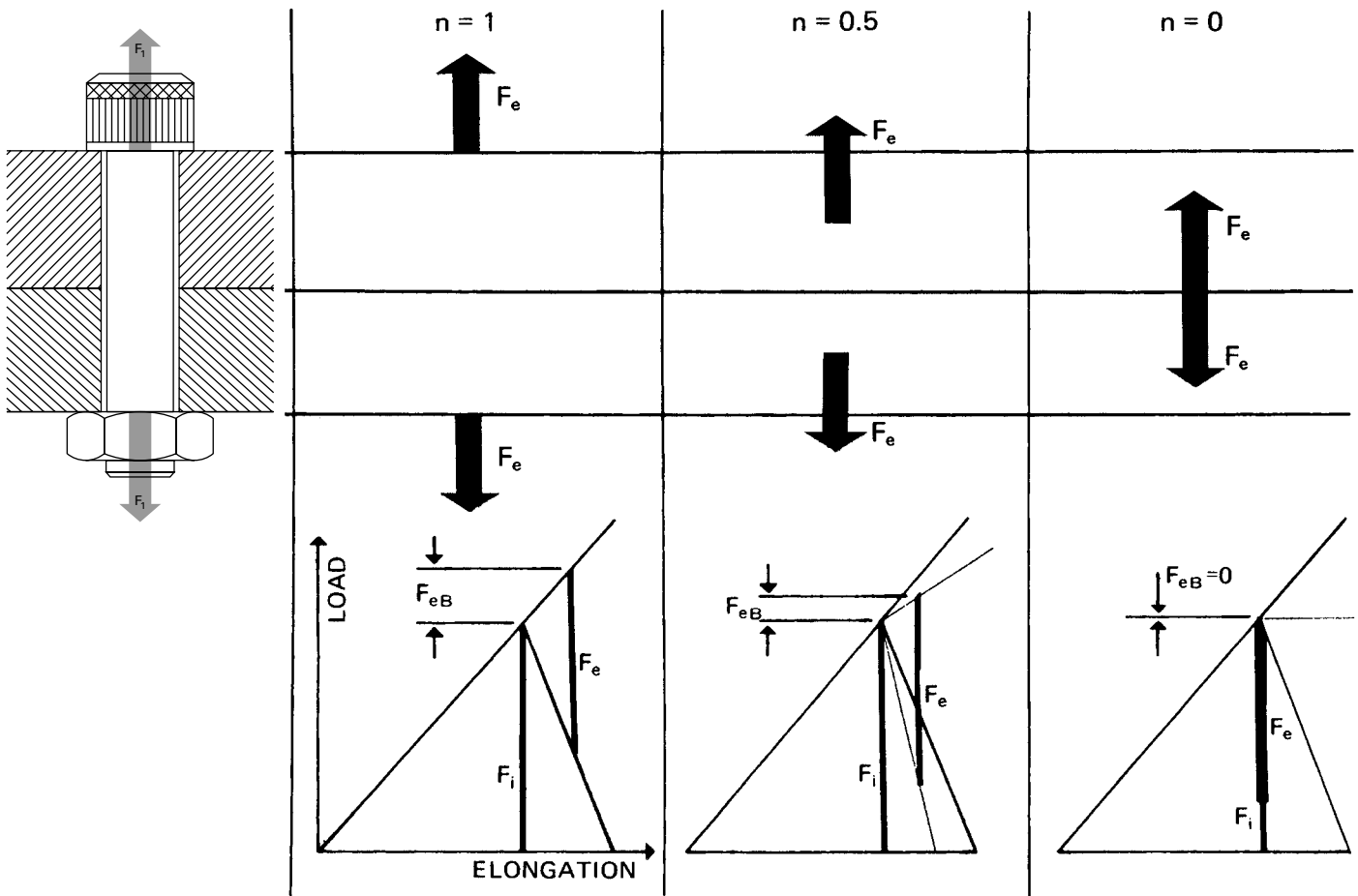


Fig. 10 Force diagrams show the effect of the loading planes of the external load on the bolt load.

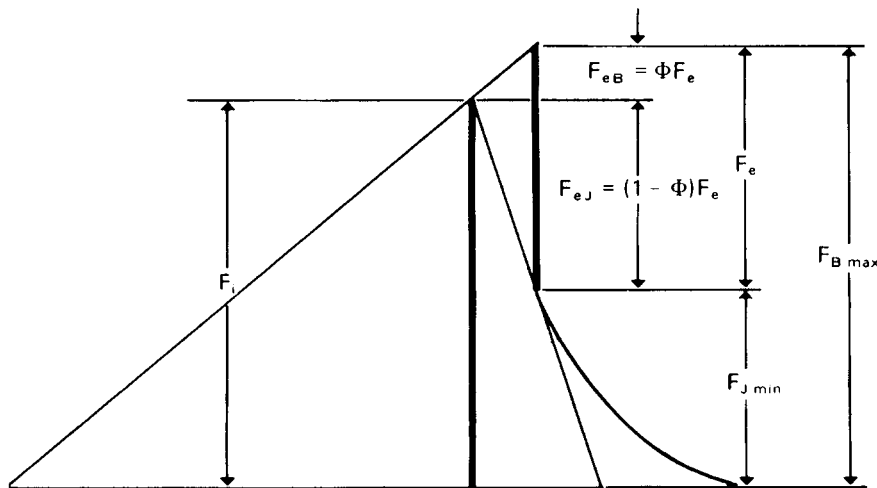


Fig. 11 Modified joint diagram shows nonlinear compression of joint at low preloads.

$F_{eB} = \Phi F_e$ must be modified to :

$$F_{eB} = n \Phi F_e$$

where n equals the ratio of the length of the clamped parts due to F_e to the joint length l_j . The value of n can range from 1, when F_e is applied under the head and nut, to 0, when F_e is applied at the joint center. Consequently the stress amplitude:

$$\sigma_B = \pm \frac{\Phi F_e}{2 A_m} \quad \text{becomes}$$

$$\sigma_B = \pm \frac{n \Phi F_e}{2 A_m}$$

General Design Formulae

Hitherto, construction of the joint diagram has assumed linear resilience of both bolt and joint members. However, recent investigations have shown that this assumption is not quite true for compressed parts.

Taking these investigations into account, the joint diagram is modified to Fig. 11. The lower portion of the joint spring rate is nonlinear, and the length of the linear portion depends on the preload level F_i . The higher F_i the longer the linear portion. By choosing a sufficiently high minimum load, $F_{i \min} > 2F_{eB}$, the non-linear range of the joint spring rate is avoided and a linear relationship between F_{eB} and F_e is maintained.

Also from Fig. 11 this formula is derived:

$$F_{i \min} = F_{J \min} + (1 - \Phi) F_e + \Delta F_i$$

where ΔF_i is the amount of preload loss to be expected. For a properly designed joint, a preload loss $\Delta F_i = - (0.005 \text{ to } 0.10) F_i$ should be expected.

The fluctuation in bolt load that results from tightening is expressed by the ratio:

$$a = \frac{F_{i \max}}{F_{i \min}}$$

where a varies between 1.25 and 3.0 depending on the tightening method.

Considering the general design formulae are:

$$F_{i \text{ nom}} = F_{J \min} = (1 - \Phi) F_e$$

$$F_{i \max} = a [F_{J \min} + (1 - \Phi) F_e + \Delta F_i]$$

$$F_{B \max} = a [F_{J \min} + (1 - \Phi) F_e + \Delta F_i] + \Phi F_e$$

Conclusion

The three requirements of concentrically loaded joints that must be met for an integral bolted joint are:

1. The maximum bolt load $F_{B \max}$ must be less than the bolt yield strength.
2. If the external load is alternating, the alternating stress must be less than the bolt endurance limit to avoid fatigue failures.
3. The joint will not lose any preload due to permanent set or vibration greater than the value assumed for ΔF_i .

SYMBOLS

A	Area (in. ²)
A_m	Area of minor thread diameter (in. ²)
A_s	Area of substitute cylinder (in. ²)
A_x	Area of bolt part 1_x (in. ²)
d	Diameter of minor thread (in.)
D_c	Outside diameter of bushing (cylinder) (in.)
D_H	Diameter of Bolt head (in.)
D_h	Diameter of hole (in.)
D_J	Diameter of Joint
E	Modulus of Elasticity (psi)
F	Load (lb)
F_e	External load (lb.)
F_{eB}	Additinal Bolt Load due to external load (lb)
F_{eJ}	Reduced Joint load due to external load (lb)
F_i	Preload on Bolt and Joint (lb)
ΔF_i	Preload loss (-lb)
$F_{i \min}$	Minimum preload (lb)
$F_{i \max}$	Maximum preload (lb)
$F_{j \text{ nom}}$	Nominal preload (lb)

$F_{B \max}$	Maximum Bolt load (lb)
$F_{J \min}$	Minimum Joint load (lb)
K	Spring rate (lb/in.)
K_B	Spring rate of Bolt (lb/in.)
K_J	Spring rate of Joint (lb/in.)
K_x	Spring rate of Bolt part 1_x (lb/in.)
l	Length (in.)
Δl	Change in length (in.)
l_B	Length of Bolt (in.)
Δl_B	Bolt elongation due to F_i (in.)
l_J	Length of Joint (in.)
Δl_J	Joint compression to F_i (in.)
l_x	Length of Bolt part x (in.)
n	$\frac{\text{Length of clamped parts}}{\text{Total Joint Length}}$
α	Tightening factor
Φ	Force ratio
λ	Bolt and Joint elongation due to F_e (in.)
σ_B	Bolt stress amplitude ($\pm$ psi)

TIGHTENING TORQUES AND THE TORQUE-TENSION RELATIONSHIP

All of the analysis and design work done in advance will have little meaning if the proper preload is not achieved. Several discussions in this technical section stress the importance of preload to maintaining joint integrity. There are many methods for measuring preload (see Table 12). However, one of the least expensive techniques that provides a reasonable level of accuracy versus cost is by measuring torque. The fundamental characteristic required is to know the relationship between torque and tension for any particular bolted joint. Once the desired design preload must be identified and specified first, *then* the torque required to induce that preload is determined.

Within the elastic range, before permanent stretch is induced, the relationship between torque and tension is essentially linear (see figure 13). Some studies have found up to 75 variables have an effect on this relationship: materials, temperature, rate of installation, thread helix angle, coefficients of friction, etc. One way that has been developed to reduce the complexity is to depend on empirical test results. That is, to perform experiments under the application conditions by measuring the induced torque and recording the resulting tension. This can be done with relatively simple, calibrated hydraulic pressure sensors, electric strain gages, or piezoelectric load cells. Once the data is gathered and plotted on a chart, the slope of the curve can be used to calculate a correlation factor. This technique has created an accepted formula for relating torque to tension.

$$T = K \times D \times P$$

T = torque, lbf.-in.

D = fastener nominal diameter, inches

P = preload, lbf.

K = "nut factor," "tightening factor," or "k-value"

If the preload and fastener diameter are selected in the design process, and the K-value for the application conditions is known, then the necessary torque can be calculated. It is noted that even with a specified torque, actual conditions at the time of installation can result in variations in the actual preload achieved (see Table 12).

One of the most critical criteria is the selection of the K-value. Accepted nominal values for many industrial applications are:

K = 0.20 for as-received steel bolts into steel holes

K = 0.15 steel bolts with cadmium plating, which acts like a lubricant,

K = 0.28 steel bolts with zinc plating.

The K-value is not the coefficient of the friction (μ); it is an empirically derived correlation factor.

It is readily apparent that if the torque intended for a zinc plated fastener is used for cadmium plated fastener, the preload will be almost two times that intended; it may actually cause the bolt to break.

Another influence is where friction occurs. For steel bolts holes, approximately 50% of the installation torque is consumed by friction under the head, 35% by thread friction, and only the remaining 15% inducing preload tension. Therefore, if lubricant is applied just on the

fastener underhead, full friction reduction will not be achieved. Similarly, if the material against which the fastener is bearing, e.g. aluminum, is different than the internal thread material, e.g. cast iron, the effective friction may be difficult to predict. These examples illustrate the importance and the value of identifying the torque-tension relationship. It is a recommend practice to contact the lubricant manufacturer for K-value information if a lubricant will be used.

The recommended seating torques for Unbrako headed socket screws are based on inducing preloads reasonably expected in practice for each type. The values for Unbrako metric fasteners are calculated using VDI2230, a complex method utilized extensively in Europe. All values assume use in the received condition in steel holes. It is understandable the designer may need preloads higher than those listed. The following discussion is presented for those cases.

TORSION-TENSION YIELD AND TENSION CAPABILITY AFTER TORQUING

Once a headed fastener has been seated against a bearing surface, the inducement of torque will be translated into both torsion and tension stresses. These stresses combine to induce twist. If torque continues to be induced, the stress along the angle of twist will be the largest stress *while the bolt is being torqued*. Consequently, the stress along the bolt axis (axial tension) will be something less. This is why a bolt can fail at a lower tensile stress *during installation* than when it is pulled in straight tension alone, eg. a tensile test. Research has indicated the axial tension can range from 135,000 to 145,000 PSI for industry socket head cap screws at torsion-tension yield, depending on diameter. Including the preload variation that can occur with various installation techniques, eg. up to 25%, it can be understood why some recommended torques induce preload reasonably lower than the yield point.

Figure 13 also illustrates the effect of straight tension applied after installation has stopped. Immediately after stopping the installation procedure there will be some relaxation, and the torsion component will drop toward zero. This leaves only the axial tension, which keeps the joint clamped together. Once the torsion is relieved, the axial tension yield value and ultimate value for the fastener will be appropriate.

Table 12
Industrial Fasteners Institute's
Torque-Measuring Method

Preload Measuring Method	Accuracy Percent	Relative Cost
Feel (operator's judgement)	±35	1
Torque wrench	±25	1.5
Turn of the nut	±15	3
Load-indicating washers	±10	7
Fastener elongation	±3 to 5	15
Strain gages	±1	20

THE TORQUE-TENSION RELATIONSHIP

Fig. 14

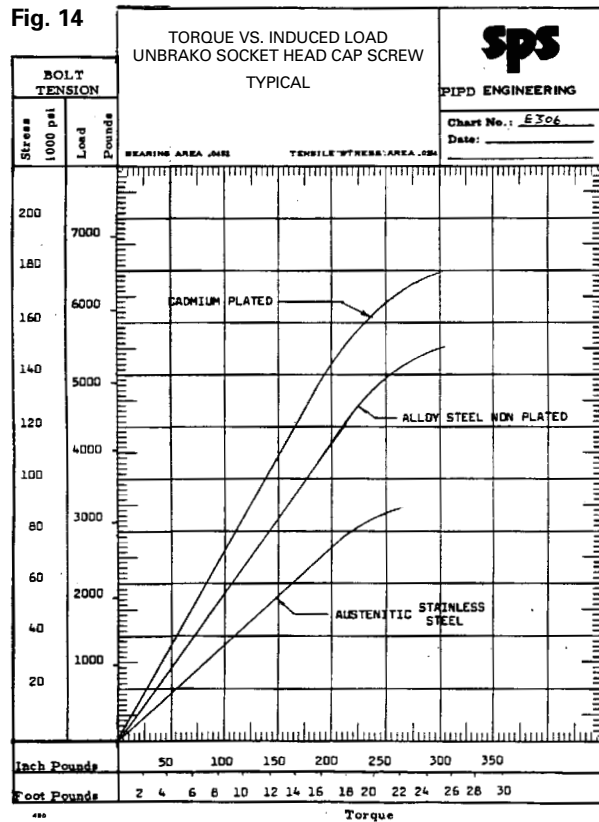


Fig. 13 Torque/Tension Relationship

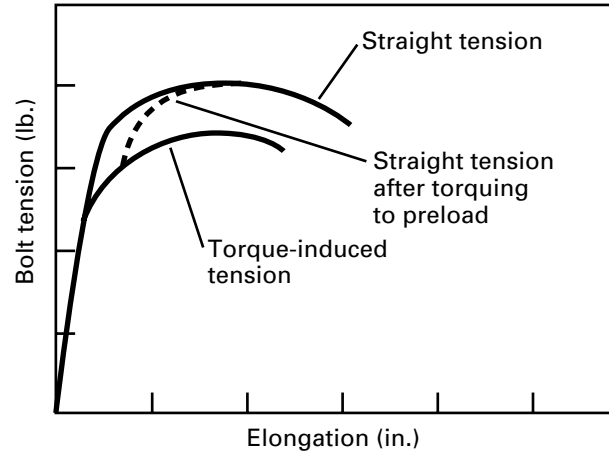


Fig. 15 Recommended Seating Torques (Inch-Lb.) for Application in Various Materials
UNBRAKO pHd (1960 Series) Socket Head Cap Screws

screw size	mild steel Rb 87 cast iron Rb 83 note 1		brass Rb 72 note 2		aluminum Rb 72 (2024-T4) note 3	
	UNC	UNF	UNC	UNF	UNC	UNF
	plain	plain	plain	plain	plain	plain
#0	—	*2.1	—	*2.1	—	*2.1
#1	*3.8	*4.1	*3.8	*4.1	*3.8	*4.1
#2	*6.3	*6.8	*6.3	*6.8	*6.3	*6.8
#3	*9.6	*10.3	*9.6	*10.3	*9.6	*10.3
#4	*13.5	*14.8	*13.5	*14.8	*13.5	*14.8
#5	*20	*21	*20	*21	*20	*21
#6	*25	*28	*25	*28	*25	*28
#8	*46	*48	*46	*48	*46	*48
#10	*67	*76	*67	*76	*67	*76
1/4	*158	*180	136	136	113	113
5/16	*326	*360	228	228	190	190
3/8	*580	635	476	476	397	397
7/16	*930	*1,040	680	680	570	570
1/2	*1,420	*1,590	1,230	1,230	1,030	1,030
9/16	*2,040	2,250	1,690	1,690	1,410	1,410
5/8	*2,820	3,120	2,340	2,340	1,950	1,950
3/4	*5,000	5,340	4,000	4,000	3,340	3,340
7/8	*8,060	8,370	6,280	6,280	5,230	5,230
1	*12,100	12,800	9,600	9,600	8,000	8,000
1 1/8	*13,800	*15,400	13,700	13,700	11,400	11,400
1 1/4	*19,200	*21,600	18,900	18,900	15,800	15,800
1 3/8	*25,200	*28,800	24,200	24,200	20,100	20,100
1 1/2	*33,600	*36,100	32,900	32,900	27,400	27,400

NOTES:

1. Torques based on 80,000 psi bearing stress under head of screw.
2. Torques based on 60,000 psi bearing stress under head of screw.
3. Torques based on 50,000 psi bearing stress under head of screw.

*Denotes torques based on 100,000 psi tensile stress in screw threads up to 1" dia., and 80,000 psi for sizes 1 1/8" dia. and larger.

To convert inch-pounds to inch-ounces – multiply by 16.

To convert inch-pounds to foot-pounds – divide by 12.

STRIPPING STRENGTH OF TAPPED HOLES

Charts and sample problems for obtaining minimum thread engagement based on applied load, material, type of thread and bolt diameter.

Knowledge of the thread stripping strength of tapped holes is necessary to develop full tensile strength of the bolt or, for that matter, the minimum engagement needed for any lesser load.

Conversely, if only limited length of engagement is available, the data help determine the maximum load that can be safely applied without stripping the threads of the tapped hole.

Attempts to compute lengths of engagement and related factors by formula have not been entirely satisfactory—mainly because of subtle differences between various materials. Therefore, strength data has been empirically developed from a series of tensile tests of tapped specimens for seven commonly used metals including steel, aluminum, brass and cast iron.

The design data is summarized in the six accompanying charts, (Charts E504-E509), and covers a range of screw thread sizes from #0 to one inch in diameter for both coarse and fine threads. Though developed from tests of Unbrako socket head cap screws having minimum ultimate tensile strengths (depending on the diameter) from 190,000 to 180,000 psi, these stripping strength values are valid for all other screws or bolts of equal or lower strength having a standard thread form. Data are based on static loading only.

In the test program, bolts threaded into tapped specimens of the metal under study were stressed in tension until the threads stripped. Load at which stripping occurred and the length of engagement of the specimen were noted. Conditions of the tests, all of which are met in a majority of industrial bolt applications, were:

- Tapped holes had a basic thread depth within the range of 65 to 80 per cent. Threads of tapped holes were Class 2B fit or better.
- Minimum amount of metal surrounding the tapped hole was 2 1/2 times the major diameter.
- Test loads were applied slowly in tension to screws having standard Class 3A threads. (Data, though, will be equally applicable to Class 2A external threads as well.)
- Study of the test results revealed certain factors that greatly simplified the compilation of thread stripping strength data:
- Stripping strengths are almost identical for loads applied either by pure tension or by screw torsion. Thus data are equally valid for either condition of application.

- Stripping strength values vary with diameter of screw. For a given load and material, larger diameter bolts required greater engagement.
- Minimum length of engagement (as a percent of screw diameter) is a straight line function of load. This permits easy interpolation of test data for any intermediate load condition.
- When engagement is plotted as a percentage of bolt diameter, it is apparent that stripping strengths for a wide range of screw sizes are close enough to be grouped in a single curve. Thus, in the accompanying charts, data for sizes #0 through #12 have been represented by a single set of curves.

With these curves, it becomes a simple matter to determine stripping strengths and lengths of engagement for any condition of application. A few examples are given below:

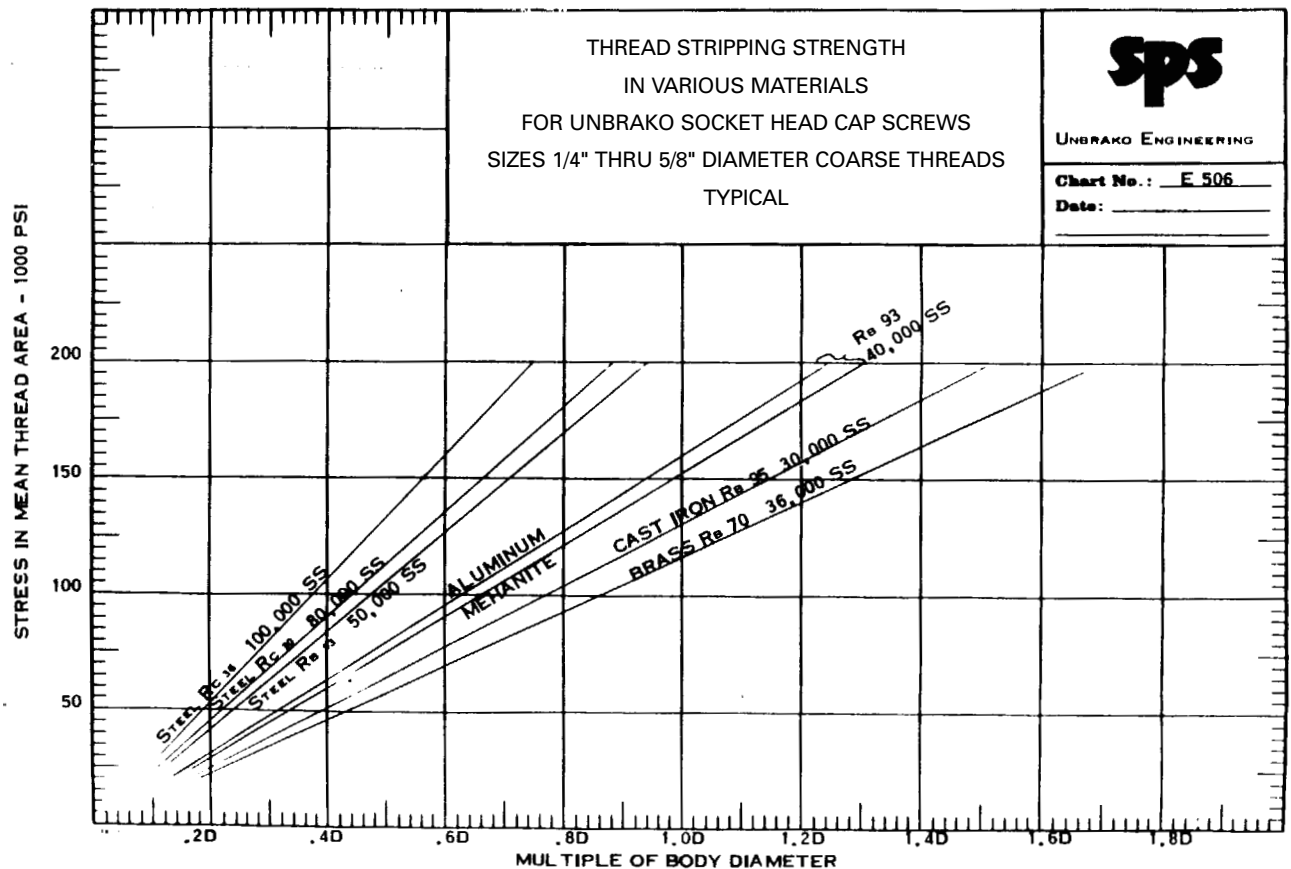
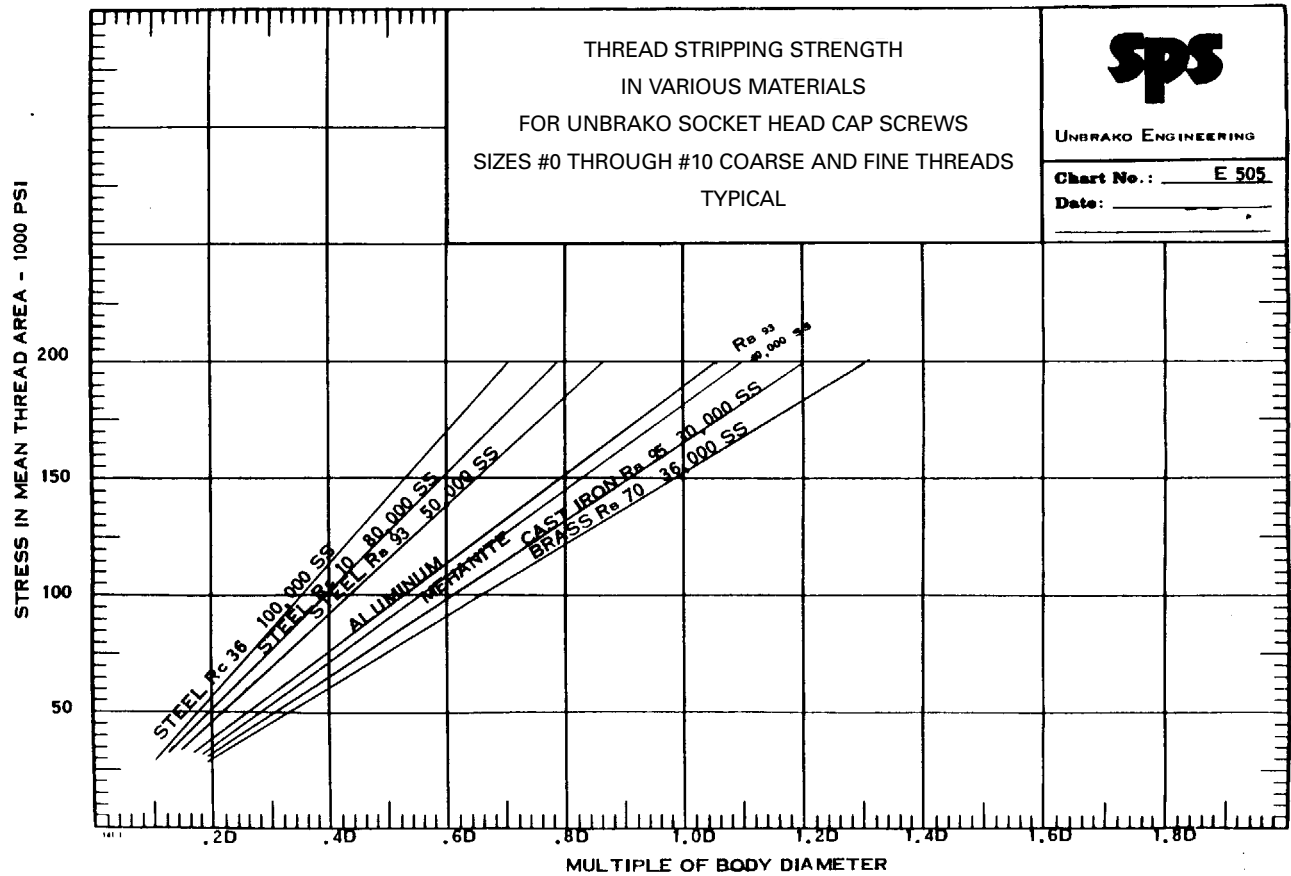
Example 1. Calculate length of thread engagement necessary to develop the minimum ultimate tensile strength (190,000 psi) of a 1/2–13 (National Coarse) Unbrako cap screw in cast iron having an ultimate shear strength of 30,000 psi. E505 is for screw sizes from #0 through #10; E506 and E507 for sizes from 1/4 in. through 5/8 in.; E508 and E509 for sizes from 3/4 in. through 1 in. Using E506 a value 1.40D is obtained. Multiplying nominal bolt diameter (0.500 in.) by 1.40 gives a minimum length of engagement of 0.700 in.

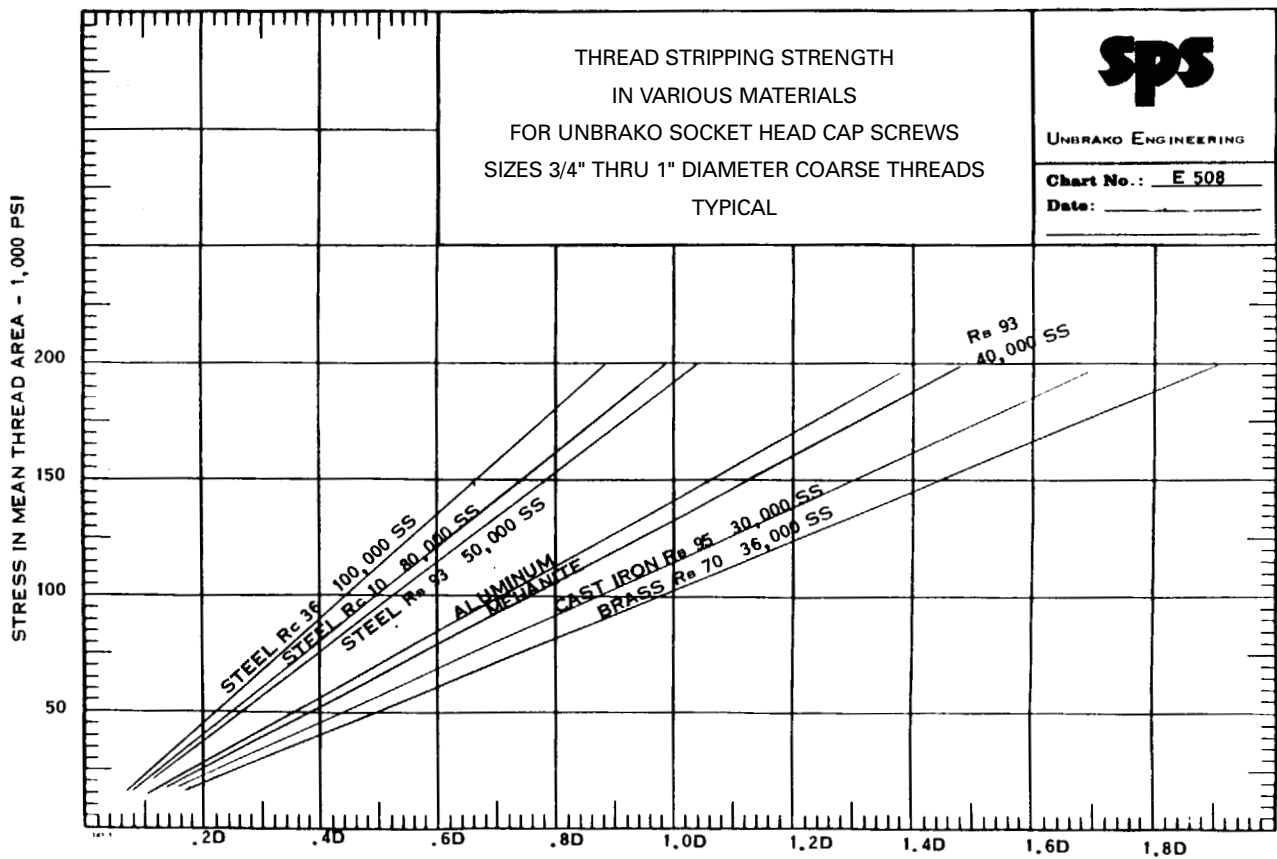
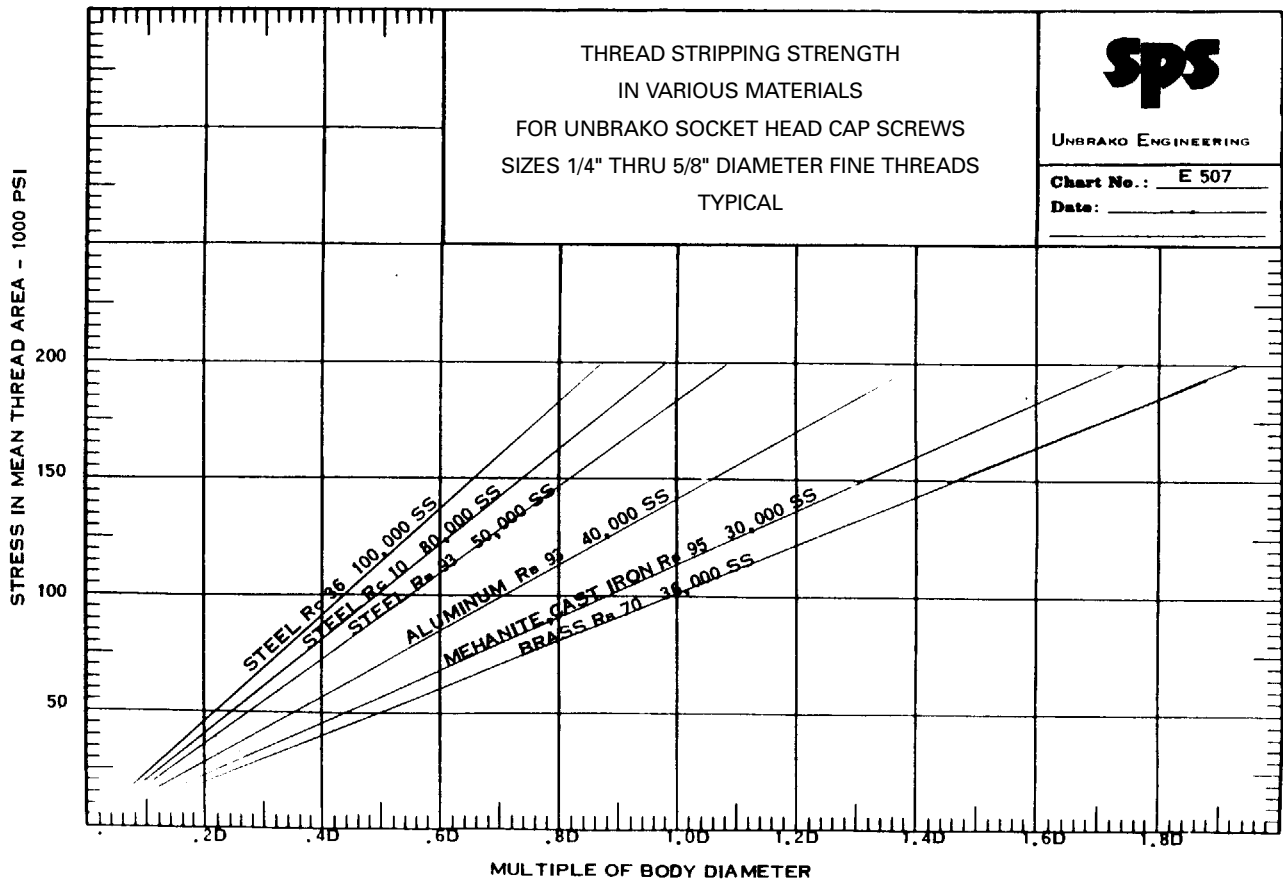
Example 2. Calculate the length of engagement for the above conditions if only 140,000 psi is to be applied. (This is the same as using a bolt with a maximum tensile strength of 140,000psi.) From E506 obtain value of 1.06D. Minimum length of engagement = (0.500) (1.06) = 0.530.

Example 3. Suppose in Example 1 that minimum length of engagement to develop full tensile strength was not available because the thickness of metal allowed a tapped hole of only 0.600 in. Hole depth in terms of bolt dia. = $0.600/0.500 = 1.20D$. By working backwards in Chart E506, maximum load that can be carried is approximately 159,000 psi.

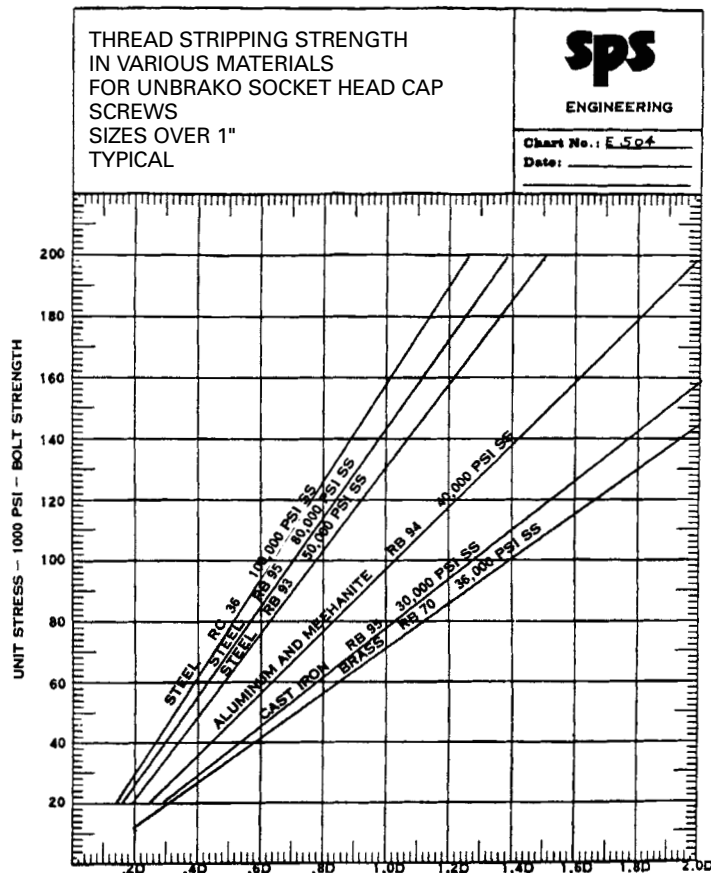
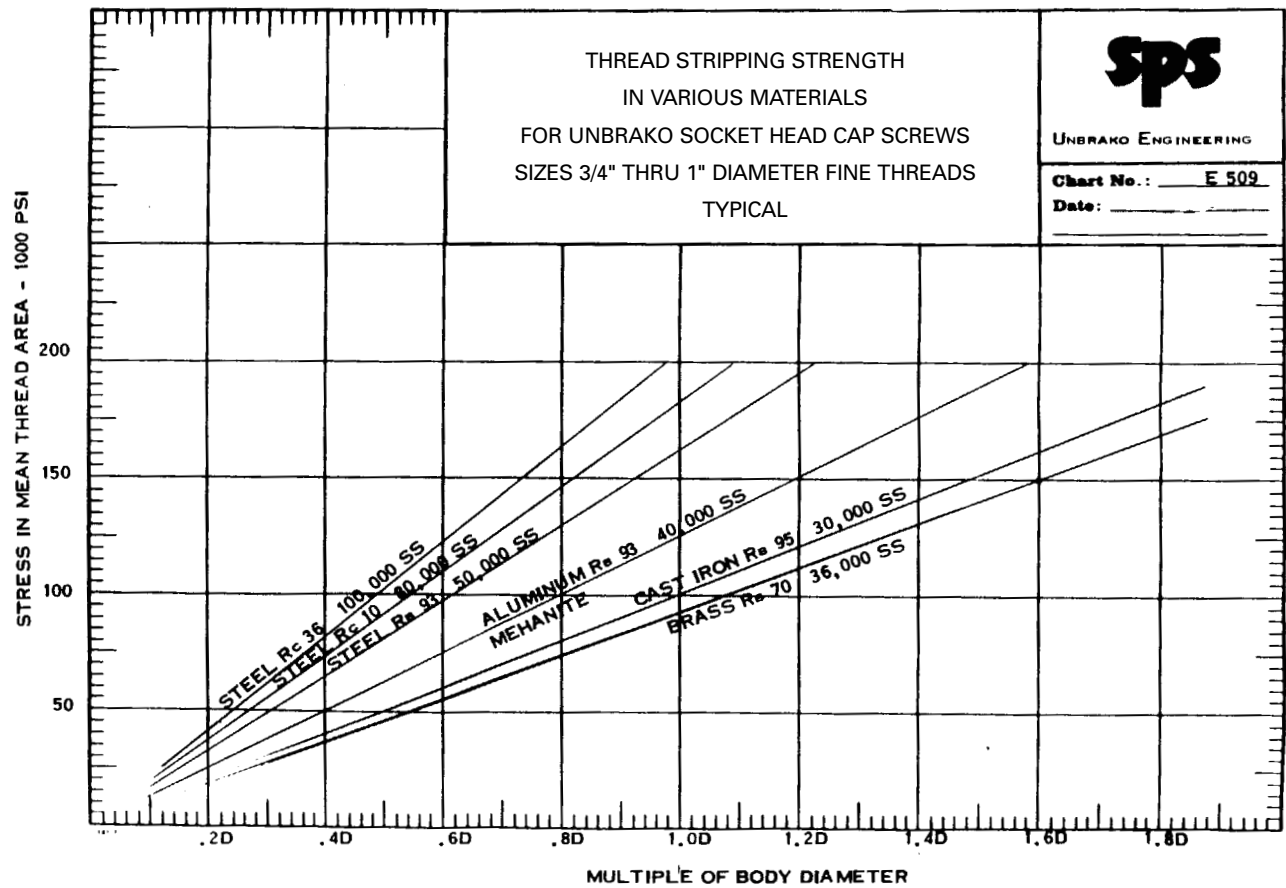
Example 4. Suppose that the hole in Example 1 is to be tapped in steel having an ultimate shear strength 65,000 psi. There is no curve for this steel in E506 but a design value can be obtained by taking a point midway between curves for the 80,000 psi and 50,000 psi steels that are listed. Under the conditions of the example, a length of engagement of 0.825D or 0.413 in. will be obtained.

STRIPPING STRENGTH OF TAPPED HOLES





STRIPPING STRENGTH OF TAPPED HOLES



HIGH-TEMPERATURE JOINTS

Bolted joints subjected to cyclic loading perform best if an initial preload is applied. The induced stress minimizes the external load sensed by the bolt, and reduces the chance of fatigue failure. At high temperature, the induced load will change, and this can adversely affect the fastener performance. It is therefore necessary to compensate for high-temperature conditions when assembling the joint at room temperature. This article describes the factors which must be considered and illustrates how a high-temperature bolted joint is designed.

In high-temperature joints, adequate clamping force or preload must be maintained in spite of temperature-induced dimensional changes of the fastener relative to the joint members. The change in preload at any given temperature for a given time can be calculated, and the effect compensated for by proper fastener selection and initial preload.

Three principal factors tend to alter the initial clamping force in a joint at elevated temperatures, provided that the fastener material retains requisite strength at the elevated temperature. These factors are: Modulus of elasticity, coefficient of thermal expansion, and relaxation.

Modulus Of Elasticity: As temperature increases, less stress or load is needed to impart a given amount of elongation or strain to a material than at lower temperatures. This means that a fastener stretched a certain amount at room temperature to develop a given preload will exert a lower clamping force at higher temperature if there is no change in bolt elongation.

Coefficient of Expansion: With most materials, the size of the part increases as the temperature increases. In a joint, both the structure and the fastener grow with an increase in temperature, and this can result, depending on the materials, in an increase or decrease in the clamping force. Thus, matching of materials in joint design can assure sufficient clamping force at both room and elevated temperatures. Table 16 lists mean coefficient of thermal expansion of certain fastener alloys at several temperatures.

Relaxation: At elevated temperatures, a material subjected to constant stress below its yield strength will flow plastically and permanently change size. This phenomenon is called creep. In a joint at elevated temperature, a fastener with a fixed distance between the bearing surface of the head and nut will produce less and less clamping force with time. This characteristic is called relaxation. It differs from creep in that stress changes while elongation or strain remains constant. Such elements as material, temperature, initial stress, manufacturing method, and design affect the rate of relaxation.

Relaxation is the most important of the three factors. It is also the most critical consideration in design of elevated-temperature fasteners. A bolted joint at 1200°F can lose as much as 35 per cent of preload. Failure to compensate for this could lead to fatigue failure through a loose joint even though the bolt was properly tightened initially.

If the coefficient of expansion of the bolt is greater than that of the joined material, a predictable amount of clamping force will be lost as temperature increases. Conversely, if the coefficient of the joined material is greater, the bolt may be stressed beyond its yield or even fracture strength. Or, cyclic thermal stressing may lead to thermal fatigue failure.

Changes in the modulus of elasticity of metals with increasing temperature must be anticipated, calculated, and compensated for in joint design. Unlike the coefficient of expansion, the effect of change in modulus is to reduce clamping force whether or not bolt and structure are the same material, and is strictly a function of the bolt metal.

Since the temperature environment and the materials of the structure are normally "fixed," the design objective is to select a bolt material that will give the desired clamping force at all critical points in the operating range of the joint. To do this, it is necessary to balance out the three factors—relaxation, thermal expansion, and modulus—with a fourth, the amount of initial tightening or clamping force.

In actual joint design the determination of clamping force must be considered with other design factors such as ultimate tensile, shear, and fatigue strength of the fastener at elevated temperature. As temperature increases the inherent strength of the material decreases. Therefore, it is important to select a fastener material which has sufficient strength at maximum service temperature.

Example

The design approach to the problem of maintaining satisfactory elevated-temperature clamping force in a joint can be illustrated by an example. The example chosen is complex but typical. A cut-and-try process is used to select the right bolt material and size for a given design load under a fixed set of operating loads and environmental conditions, Fig. 17.

The first step is to determine the change in thickness, Δt , of the structure from room to maximum operating temperature.

For the AISI 4340 material:

$$\Delta t_1 = t_1(T_2 - T_1)\alpha$$

$$\Delta t_1 = (0.50)(800 - 70)(7.4 \times 10^{-6})$$

$$\Delta t_1 = 0.002701 \text{ in.}$$

For the AMS 6304 material:

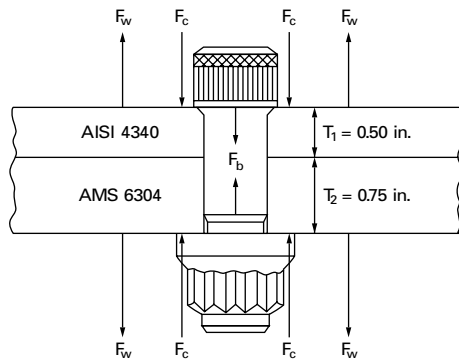
$$\Delta t_2 = (0.75)(800 - 70)(7.6 \times 10^{-6})$$

$$\Delta t_2 = 0.004161 \text{ in.}$$

The total increase in thickness for the joint members is 0.00686 in.

The total effective bolt length equals the total joint thickness plus one-third of the threads engaged by the nut. If it is assumed that the smallest diameter bolt should be used for weight saving, then a 1/4-in. bolt should be tried. Thread engagement is approximately one diameter, and the effective bolt length is:

HIGH-TEMPERATURE JOINTS



- d = Bolt diam, in.
 E = Modulus of elasticity, psi
 F_b = Bolt preload, lb
 F_c = Clamping force, lb ($F_b = F_c$)
 F_w = Working load = 1500 lb static + 100 lb cyclic
 L = Effective bolt length, in.
 T_1 = Room temperature = 70°F
 T_2 = Maximum operating temperature for 1000 hr = 800°F
 t = Panel thickness, in.
 a = Coefficient of thermal expansion

Fig. 17 – Parameters for joint operating at 800°F.

$$L = t_1 + t_2 + (1/3)d$$

$$L = 0.50 + 0.75 + (1/3 \times 0.25)$$

$$L = 1.333 \text{ in.}$$

The ideal coefficient of thermal expansion of the bolt material is found by dividing the total change in joint thickness by the bolt length times the change in temperature.

$$\alpha_b = \frac{\Delta t}{L \times \Delta t}$$

$$\alpha = \frac{.00686}{(1.333)(800 - 70)} = 7.05 \times 10^{-6} \text{ in./in./deg. F}$$

The material, with the nearest coefficient of expansion is with a value of 9,600,000 at 800°F.

To determine if the bolt material has sufficient strength and resistance to fatigue, it is necessary to calculate the stress in the fastener at maximum and minimum load. The bolt load plus the cyclic load divided by the tensile stress of the threads will give the maximum stress. For a 1/4-28 bolt, tensile stress area, from thread handbook H 28, is 0.03637 sq. in. The maximum stress is

$$S_{max} = \frac{\text{Bolt load}}{\text{Stress area}} = \frac{1500 + 100}{0.03637}$$

$$S_{max} = 44,000 \text{ psi}$$

and the minimum bolt stress is 41,200 psi.

H-11 has a yield strength of 175,000 psi at 800°F, Table 3, and therefore should be adequate for the working loads.

A Goodman diagram, Fig. 18, shows the extremes of stress within which the H-11 fastener will not fail by fatigue. At the maximum calculated load of 44,000 psi, the fastener will withstand a minimum cyclic loading at 800°F of about 21,000 psi without fatigue failure.

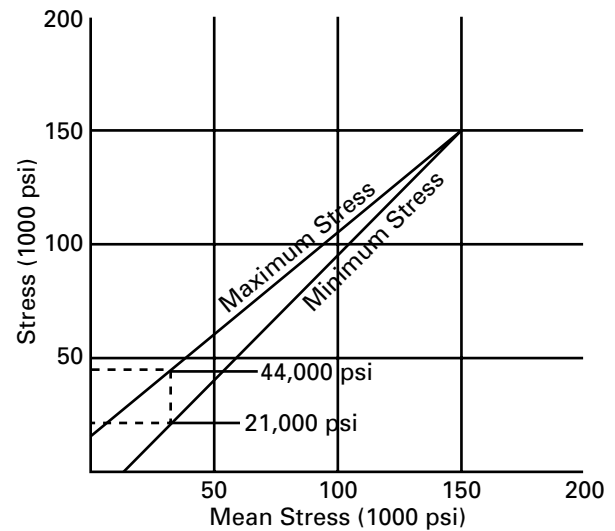


Fig. 18 – Goodman diagram of maximum and minimum operating limits for H-11 fastener at 800°F. Bolts stressed within these limits will give infinite fatigue life.

Because of relaxation, it is necessary to determine the initial preload required to insure 1500-lb. clamping force in the joint after 1000 hr at 800°F.

When relaxation is considered, it is necessary to calculate the maximum stress to which the fastener is subjected. Because this stress is not constant in dynamic joints, the resultant values tend to be conservative. Therefore, a maximum stress of 44,000 psi should be considered although the necessary stress at 800°F need be only 41,200 psi. Relaxation at 44,000 psi can be interpolated from the figure, although an actual curve could be constructed from tests made on the fastener at the specific conditions.

The initial stress required to insure a clamping stress of 44,000 psi after 1000 hr at 800°F can be calculated by interpolation.

$$x = 61,000 - 44,000 = 17,000$$

$$y = 61,000 - 34,000 = 27,000$$

$$B = 80,000 - 50,000 = 30,000$$

$$A = 80,000 - C$$

$$\frac{x}{y} = \frac{A}{B} \quad \frac{17,000}{27,000} = \frac{80,000 - C}{30,000}$$

$$C = 61,100 \text{ psi}$$

The bolt elongation required at this temperature is calculated by dividing the stress by the modulus at temperature and multiplying by the effective length of the bolt. That is: $(61,000 \times 1.333) / 24.6 \times 10^6 = 0.0033$

Since the joint must be constructed at room temperature, it is necessary to determine the stresses at this state. Because the modulus of the fastener material changes with temperature, the clamping force at room temperature will not be the same as at 800°F. To deter-

mine the clamping stress at assembly conditions, the elongation should be multiplied by the modulus of elasticity at room temperature.

$$.0033 \times 30.6 \times 10^6 = 101,145 \text{ psi}$$

The assembly conditions will be affected by the difference between the ideal and actual coefficients of expansion of the joint. The ideal coefficient for the fastener material was calculated to be 7.05 but the closest material – H-11 – has a coefficient of 7.1. Since this material has a greater expansion than calculated, there will be a reduction in clamping force resulting from the increase in temperature. This amount equals the difference between the ideal and the actual coefficients multiplied by the change in temperature, the length of the fastener, and the modulus of elasticity at 70°F.

$$[(7.1 - 7.05) \times 10^{-6}][800 - 70][1.333] \times$$

$$[30.6 \times 10^6] = 1,490 \text{ psi}$$

The result must be added to the initial calculated stresses to establish the minimum required clamping stress needed for assembling the joint at room temperature.

$$101,145 + 1,490 = 102,635 \text{ psi}$$

Finally, the method of determining the clamping force or preload will affect the final stress in the joint at operating conditions. For example, if a torque wrench is

used to apply preload (the most common and simplest method available), a plus or minus 25 per cent variation in induced load can result. Therefore, the maximum load which could be expected in this case would be 1.5 times the minimum, or:

$$(1.5)(102,635) = 153,950 \text{ psi}$$

This value does not exceed the room-temperature yield strength for H-11 given in Table 19.

Since there is a decrease in the clamping force with an increase in temperature and since the stress at operating temperature can be higher than originally calculated because of variations in induced load, it is necessary to ascertain if yield strength at 800°F will be exceeded

$$\frac{(\text{max stress at } 70^\circ\text{F} + \text{change in stress}) \times E \text{ at } 800^\circ\text{F}}{E \text{ at } 70^\circ\text{F}}$$

$$\frac{[153,950 + (-1490)] \times 24.6 \times 10^6}{30.6 \times 10^6} = 122,565$$

This value is less than the yield strength for H-11 at 800°F, Table 19. Therefore, a 1/4-28 H-11 bolt stressed between 102,635 psi and 153,950 psi at room temperature will maintain a clamping load 1500 lb at 800°F after 1000 hr of operation. A cyclic loading of 100 lb, which results in a bolt loading between 1500 and 1600 lb will not cause fatigue failure at the operating conditions.

Table 16

PHYSICAL PROPERTIES OF MATERIALS USED TO MANUFACTURE ALLOY STEEL SHCS'S

Coefficient of Thermal Expansion, $\mu\text{m/m}/^\circ\text{K}^1$

20°C to 68°F to	100 212	200 392	300 572	400 752	500 932	600 1112
Material						
5137M, 51B37M ²	–	12.6	13.4	13.9	14.3	14.6
4137 ³	11.2	11.8	12.4	13.0	13.6	–
4140 ³	12.3	12.7	–	13.7	–	14.5
4340 ³	–	12.4	–	13.6	–	14.5
8735 ³	11.7	12.2	12.8	13.5	–	14.1
8740 ³	11.6	12.2	12.8	13.5	–	14.1

Modulus of Elongation (Young's Modulus)

$$E = 30,000,000 \text{ PSI/in/in}$$

NOTES:

1. Developed from ASM, Metals HDBK, 9th Edition, Vol. 1 ($^\circ\text{C} = ^\circ\text{K}$ for values listed)
2. ASME SA574
3. AISI
4. Multiply values in table by .556 for $\mu\text{in/in}/^\circ\text{F}$.

Table 19 - Yield Strength at Various Temperatures

Alloy	Temperature (F)			
	70	800	1000	1200
Stainless Steels				
Type 302	35,000	35,000	34,000	30,000
Type 403	145,000	110,000	95,000	38,000
PH 15-7 Mo	220,000	149,000	101,000	–
High Strength Iron-Base Stainless Alloys				
A 286	95,000	95,000	90,000	85,000
AMS 5616	113,000	80,000	60,000	40,000
Unitemp 212	150,000	140,000	135,000	130,000
High Strength Iron-Base Alloys				
AISI 4340	200,000	130,000	75,000	–
H-11 (AMS 6485)	215,000	175,000	155,000	–
AMS 6340	160,000	100,000	75,000	–
Nickel-Base Alloys				
Inconel X	115,000	–	–	98,000
Waspaloy	115,000	–	106,000	100,000

CORROSION IN THREADED FASTENERS

All fastened joints are, to some extent, subjected to corrosion of some form during normal service life. Design of a joint to prevent premature failure due to corrosion must include considerations of the environment, conditions of loading, and the various methods of protecting the fastener and joint from corrosion.

Three ways to protect against corrosion are:

1. Select corrosion-resistant material for the fastener.
2. Specify protective coatings for fastener, joint interfaces, or both.
3. Design the joint to minimize corrosion.

The solution to a specific corrosion problem may require using one or all of these methods. Economics often necessitate a compromise solution.

Fastener Material

The use of a suitably corrosion-resistant material is often the first line of defense against corrosion. In fastener design, however, material choice may be only one of several important considerations. For example, the most corrosion-resistant material for a particular environment may just not make a suitable fastener.

Basic factors affecting the choice of corrosion resistant threaded fasteners are:

- Tensile and fatigue strength.
- Position on the galvanic series scale of the fastener and materials to be joined.
- Special design considerations: Need for minimum weight or the tendency for some materials to gall.
- Susceptibility of the fastener material to other types of less obvious corrosion. For example, a selected material may minimize direct attack of a corrosive environment only to be vulnerable to fretting or stress corrosion.

Some of the more widely used corrosion-resistant materials, along with approximate fastener tensile strength ratings at room temperature and other pertinent properties, are listed in Table 1. Sometimes the nature of corrosion properties provided by these fastener materials is subject to change with application and other condi-

tions. For example, stainless steel and aluminum resist corrosion only so long as their protective oxide film remains unbroken. Alloy steel is almost never used, even under mildly corrosive conditions, without some sort of protective coating. Of course, the presence of a specific corrosive medium requires a specific corrosion-resistant fastener material, provided that design factors such as tensile and fatigue strength can be satisfied.

Protective Coating

A number of factors influence the choice of a corrosion-resistant coating for a threaded fastener. Frequently, the corrosion resistance of the coating is not a principal consideration. At times it is a case of economics. Often, less-costly fastener material will perform satisfactorily in a corrosive environment if given the proper protective coating.

Factors which affect coating choice are:

- Corrosion resistance
- Temperature limitations
- Embrittlement of base metal
- Effect on fatigue life
- Effect on locking torque
- Compatibility with adjacent material
- Dimensional changes
- Thickness and distribution
- Adhesion characteristics

Conversion Coatings: Where cost is a factor and corrosion is not severe, certain conversion-type coatings are effective. These include a black-oxide finish for alloy-steel screws and various phosphate base coatings for carbon and alloy-steel fasteners. Frequently, a rust-preventing oil is applied over a conversion coating.

Paint: Because of its thickness, paint is normally not considered for protective coatings for mating threaded fasteners. However, it is sometimes applied as a supplemental treatment at installation. In special cases, a fastener may be painted and installed wet, or the entire joint may be sealed with a coat of paint after installation.

TABLE 1 – TYPICAL PROPERTIES OF CORROSION RESISTANT FASTENER MATERIALS

Materials Stainless Steels	Tensile Strength (1000 psi)	Yield Strength at 0.2% offset (1000 psi)	Maximum Service Temp (F)	Mean Coefficient of Thermal Expan. (in./in./deg F)	Density (lbs/cu in.)	Base Cost Index	Position on Galvanic Scale
303, passive	80	40	800	10.2	0.286	Medium	8
303, passive, cold worked	125	80	800	10.3	0.286	Medium	9
410, passive	170	110	400	5.6	0.278	Low	15
431, passive	180	140	400	6.7	0.280	Medium	16
17-4 PH	200	180	600	6.3	0.282	Medium	11
17-7 PH	200	185	600	6.7	0.276	Medium	14
AM 350	200	162	800	7.2	0.282	Medium	13
15-7 Mo	200	155	600	–	0.277	Medium	12
A-286	150	85	1200	9.72	0.286	Medium	6
A-286, cold worked	220	170	1200	–	0.286	High	7

Electroplating: Two broad classes of protective electroplating are: 1. The barrier type-such as chrome plating-which sets up an impervious layer or film that is more noble and therefore more corrosion resistant than the base metal. 2. The sacrificial type, zinc for example, where the metal of the coating is less noble than the base metal of the fastener. This kind of plating corrodes sacrificially and protects the fastener.

Noble-metal coatings are generally not suitable for threaded fasteners-especially where a close-tolerance fit is involved. To be effective, a noble-metal coating must be at least 0.001 in. thick. Because of screw-thread geometry, however, such plating thickness will usually exceed the tolerance allowances on many classes of fit for screws.

Because of dimensional necessity, threaded fastener coatings, since they operate on a different principle, are effective in layers as thin as 0.0001 to 0.0002 in.

The most widely used sacrificial platings for threaded fasteners are cadmium, zinc, and tin. Frequently, the cadmium and zinc are rendered even more corrosion resistant by a post-plating chromate-type conversion treatment. Cadmium plating can be used at temperatures to 450°F. Above this limit, a nickel cadmium or nickel-zinc alloy plating is recommended. This consists of alternate deposits of the two metals which are heat-diffused into a uniform alloy coating that can be used for applications to 900°F. The alloy may also be deposited directly from the plating bath.

Fastener materials for use in the 900 to 1200°F range (stainless steel, A-286), and in the 1200° to 1800°F range (high-nickel-base super alloys) are highly corrosion resistant and normally do not require protective coatings, except under special environment conditions.

Silver plating is frequently used in the higher temperature ranges for lubrication to prevent galling and seizing, particularly on stainless steel. This plating can cause a galvanic corrosion problem, however, because of the high nobility of the silver.

Hydrogen Embrittlement: A serious problem, known as hydrogen embrittlement, can develop in plated alloy steel fasteners. Hydrogen generated during plating can diffuse into the steel and embrittle the bolt. The result is often a delayed and total mechanical failure, at tensile levels far below the theoretical strength, high-hardness structural parts are particularly susceptible to this condition. The problem can be controlled by careful selection of plating formulation, proper plating procedure, and sufficient baking to drive off any residual hydrogen.

Another form of hydrogen embrittlement, which is more difficult to control, may occur after installation. Since electrolytic cell action liberates hydrogen at the cathode, it is possible for either galvanic or concentration-cell corrosion to lead to embrittling of the bolt material.

Joint Design

Certain precautions and design procedures can be followed to prevent, or at least minimize, each of the various types of corrosion likely to attack a threaded joint. The most important of these are:

For Direct Attack: Choose the right corrosion-resistant material. Usually a material can be found that will provide the needed corrosion resistance without sacrifice of other important design requirements. Be sure that the fastener material is compatible with the materials being joined.

Corrosion resistance can be increased by using a conversion coating such as black oxide or a phosphate-base treatment. Alternatively, a sacrificial coating such as zinc plating is effective.

For an inexpensive protective coating, lacquer or paint can be used where conditions permit.

For Galvanic Corrosion: If the condition is severe, electrically insulate the bolt and joint from each other..

The fastener may be painted with zinc chromate primer prior to installation, or the entire joint can be coated with lacquer or paint.

Another protective measure is to use a bolt that is cathodic to the joint material and close to it in the galvanic series. When the joint material is anodic, corrosion will spread over the greater area of the fastened materials. Conversely, if the bolt is anodic, galvanic action is most severe.

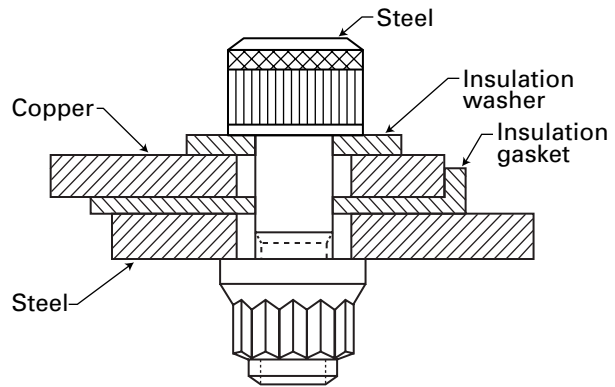


FIG. 1.1 – A method of electrically insulating a bolted joint to prevent galvanic corrosion.

For Concentration-Cell Corrosion: Keep surfaces smooth and minimize or eliminate lap joints, crevices, and seams. Surfaces should be clean and free of organic material and dirt. Air trapped under a speck of dirt on the surface of the metal may form an oxygen concentration cell and start pitting.

For maximum protection, bolts and nuts should have smooth surfaces, especially in the seating areas. Flush-head bolts should be used where possible. Further, joints can be sealed with paint or other sealant material.

For Fretting Corrosion: Apply a lubricant (usually oil) to mating surfaces. Where fretting corrosion is likely to occur: 1. Specify materials of maximum practicable hardness. 2. Use fasteners that have residual compressive stresses on the surfaces that may be under attack. 3. Specify maximum preload in the joint. A higher clamping force results in a more rigid joint with less relative movement possible between mating services.

CORROSION IN THREADED FASTENERS

For Stress Corrosion: Choose a fastener material that resists stress corrosion in the service environment. Reduce fastener hardness (if reduced strength can be tolerated), since this seems to be a factor in stress corrosion.

Minimize crevices and stress risers in the bolted joint and compensate for thermal stresses. Residual stresses resulting from sudden changes in temperature accelerate stress corrosion.

If possible, induce residual compressive stresses into the surface of the fastener by shot-peening or pressure rolling.

For Corrosion Fatigue: In general, design the joint for high fatigue life, since the principal effect of this form of corrosion is reduced fatigue performance. Factors extending fatigue performance are: 1. Application and maintenance of a high preload. 2. Proper alignment to avoid bending stresses.

If the environment is severe, periodic inspection is recommended so that partial failures may be detected before the structure is endangered.

As with stress and fretting corrosion, compressive stresses induced on the fastener surfaces by thread rolling, fillet rolling, or shot peening will reduce corrosion fatigue. Further protection is provided by surface coating.

TYPES OF CORROSION

Direct Attack...most common form of corrosion affecting all metals and structural forms. It is a direct and general chemical reaction of the metal with a corrosive medium—liquid, gas, or even a solid.

Galvanic Corrosion...occurs with dissimilar metals contact. Presence of an electrolyte, which may be nothing more than an individual atmosphere, causes corrosive action in the galvanic couple. The anodic, or less noble material, is the sacrificial element. Hence, in a joint of stainless steel and titanium, the stainless steel corrodes. One of the worst galvanic joints would consist of magnesium and titanium in contact.

Concentration Cell Corrosion...takes place with metals in close proximity and, unlike galvanic corrosion, does not require dissimilar metals. When two or more areas on the surface of a metal are exposed to different concentrations of the same solution, a difference in electrical potential results, and corrosion takes place.

If the solution consists of salts of the metal itself, a metal-ion cell is formed, and corrosion takes place on the surfaces in close contact. The corrosive solution between the two surfaces is relatively more stagnant (and thus has a higher concentration of metal ions in solution) than the corrosive solution immediately outside the crevice.

A variation of the concentration cell is the oxygen cell in which a corrosive medium, such as moist air, contains different amounts of dissolved oxygen at different points. Accelerated corrosion takes place between hidden surfaces (either under the bolt head or nut, or between bolted materials) and is likely to advance without detection.

Fretting...corrosive attack or deterioration occurring between containing, highly-loaded metal surfaces subjected to very slight (vibratory) motion. Although the mechanism is not completely understood, it is probably a highly accelerated form of oxidation under heat and stress. In threaded joints, fretting can occur between mating threads, at the bearing surfaces under the head of the screw, or under the nut. It is most likely to occur in high tensile, high-frequency, dynamic-load applications. There need be no special environment to induce this form of corrosion...merely the presence of air plus vibratory rubbing. It can even occur when only one of the materials in contact is metal.

Stress Corrosion Cracking...occurs over a period of time in high-stressed, high-strength joints. Although not fully understood, stress corrosion cracking is believed to be caused by the combined and mutually accelerating effects of static tensile stress and corrosive environment. Initial pitting somehow takes place which, in turn, further increases stress build-up. The effect is cumulative and, in a highly stressed joint, can result in sudden failure.

Corrosion Fatigue...accelerated fatigue failure occurring in the presence of a corrosive medium. It differs from stress corrosion cracking in that dynamic alternating stress, rather than static tensile stress, is the contributing agent.

Corrosion fatigue affects the normal endurance limit of the bolt. The conventional fatigue curve of a normal bolt joint levels off at its endurance limit, or maximum dynamic load that can be sustained indefinitely without fatigue failure. Under conditions of corrosion fatigue, however, the curve does not level off but continues downward to a point of failure at a finite number of stress cycles.

THE IMPACT PERFORMANCE OF THREADED FASTENERS

Much has been written regarding the significance of the notched bar impact testing of steels and other metallic materials. The Charpy and Izod type test relate notch behavior (brittleness versus ductility) by applying a single overload of stress. The results of these tests provide quantitative comparisons but are not convertible to energy values useful for engineering design calculations. The results of an individual test are related to that particular specimen size, notch geometry and testing conditions and cannot be generalized to other sizes of specimens and conditions.

The results of these tests are useful in determining the susceptibility of a material to brittle behavior when the applied stress is perpendicular to the major stress.

In externally threaded fasteners, however, the loading usually is applied in a longitudinal direction. The impact test, therefore, which should be applicable would be one where the applied impact stress supplements the major stress. Only in shear loading on fasteners is the major stress in the transverse direction.

Considerable testing has been conducted in an effort to determine if a relationship exists between the Charpy V notch properties of a material and the tension properties of an externally threaded fastener manufactured from the same material.

Some conclusions which can be drawn from the extensive impact testing are as follows:

1. The tension impact properties of externally threaded fasteners do not follow the Charpy V notch impact pattern.
2. Some of the variables which effect the tension impact properties are:
 - A. The number of exposed threads
 - B. The length of the fastener
 - C. The relationship of the fastener shank diameter to the thread area.
 - D. The hardness or fastener ultimate tensile strength

Following are charts showing tension impact versus Charpy impact properties, the effect of strength and diameter on tension impact properties and the effect of test temperature.

Please note from figure 21 that while the Charpy impact strength of socket head cap screw materials are decreasing at sub-zero temperatures, the tension impact strength of the same screws is increasing. This compares favorable with the effect of cryogenic temperatures on the tensile strength of the screws. Note the similar increase in tensile strength shown in figure 22.

It is recommended, therefore, that less importance be attached to Charpy impact properties of materials which are intended to be given to impact properties for threaded fasteners. If any consideration is to be given to impact properties of bolts or screws, it is advisable to investigate the tension impact properties of full size fasteners since this more closely approximates the actual application.

TABLE 20
LOW-TEMPERATURE IMPACT PROPERTIES OF SELECTED ALLOY STEELS

AISI no.	composition, %					heat temperature*		Hardness Rc	impact energy, ft.-lb					transition temp. (50% brittle) °F
	C	Mn	Ni	Cr	Mo	quenching temp. F+	tempering temp. F		-300°F	-200°F	-100°F	0°F	100°F	
4340	0.38	0.77	1.65	0.93	0.21	1550	400	52	11	15	20	21	21	-
							600	48	10	14	15	15	16	-
							800	44	9	13	16	21	25	-
							1000	38	15	18	28	36	36	-130
							1200	30	15	28	55	55	55	-185
4360	0.57	0.87	1.62	1.08	0.22	1475	800	48	5	6	10	11	14	-
							1000	40	9	10	13	18	23	-10
							1200	30	12	15	25	42	43	-110
4380	0.76	0.91	1.67	1.11	0.21	1450	800	49	4	5	8	9	10	-
							1000	42	8	8	10	12	15	60
							1200	31	5	11	19	33	38	-50
4620	0.20	0.67	1.85	0.30	0.18	1650	300	42	14	20	28	35	35	-
							800	34	11	16	33	55	55	-
							1000	29	16	34	55	78	78	-
							1200	19	17	48	103	115	117	-
4640	0.43	0.69	1.78	0.29	0.20	1550	800	42	16	17	20	25	27	-
							1000	37	17	22	35	39	69	-190
							1200	29	17	30	55	97	67	-180
4680	0.74	0.77	1.81	0.30	0.21	1450	800	46	5	8	13	15	16	-
							1000	41	11	12	15	19	22	-
							1200	31	11	13	17	39	43	-
8620	0.20	0.89	0.60	0.68	0.20	1650	300	43	11	16	23	35	35	-
							800	36	8	13	20	35	45	-20
							1000	29	25	33	65	76	76	-150
							1200	21	10	85	107	115	117	-195
8630	0.34	0.77	0.66	0.62	0.22	1575	800	41	7	12	17	25	31	0
							1000	34	11	20	43	53	54	-155
							1200	27	18	28	74	80	82	-165
8640	0.45	0.78	0.65	0.61	0.20	1550	800	46	5	10	14	20	23	-
							1000	38	11	15	24	40	40	-110
							1200	30	18	22	49	63	66	-140
8660	0.56	0.81	0.70	0.56	0.25	1475	800	47	4	6	10	13	16	-
							1000	41	10	12	15	20	30	-10
							1200	30	16	18	25	54	60	-90

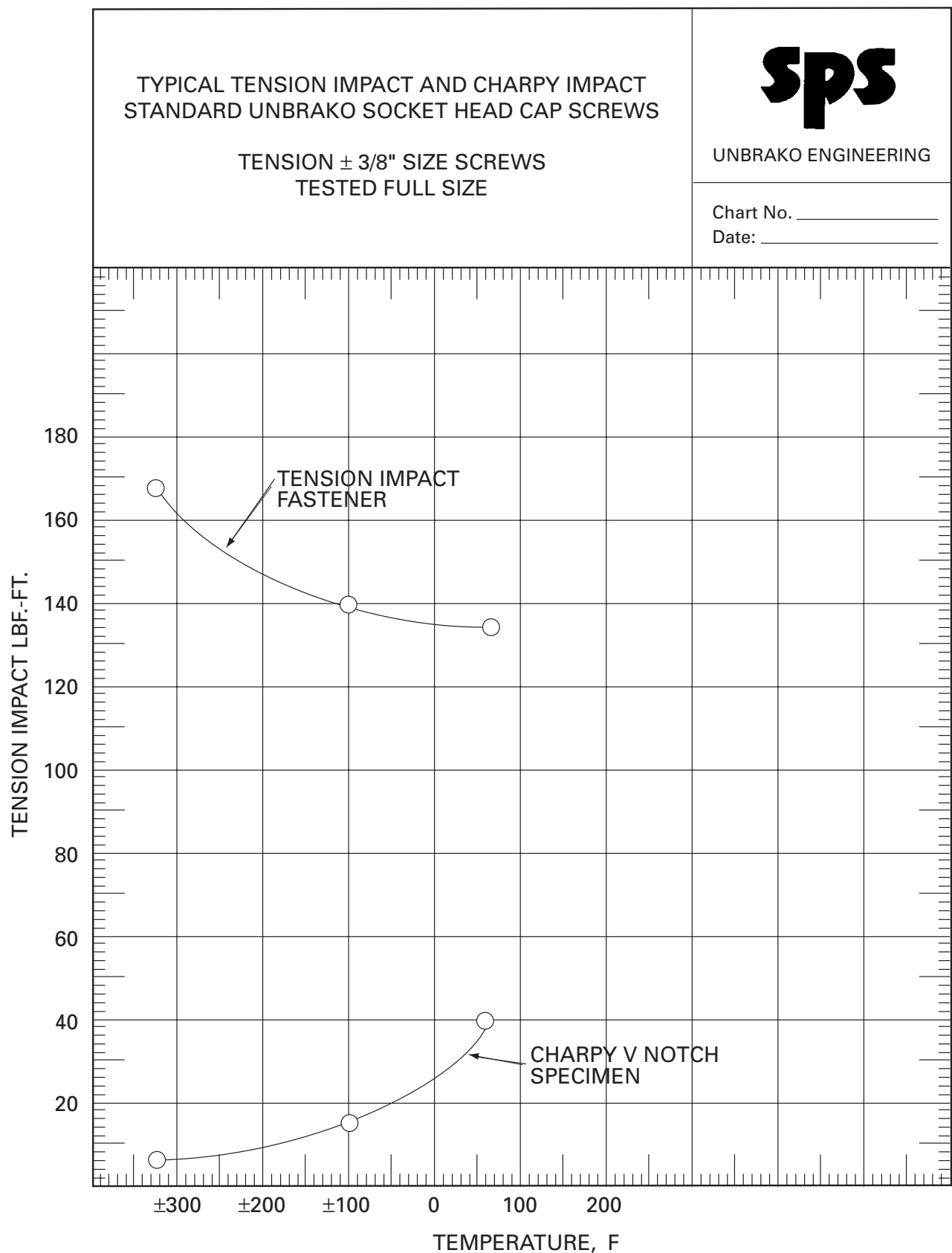


FIG. 21

TYPICAL TENSION IMPACT STRENGTH, EFFECT OF FASTENER STRENGTH AND DIAMETER

ROOM TEMPERATURE

SPS

UNBRAKO ENGINEERING

Chart No. _____

Date: _____

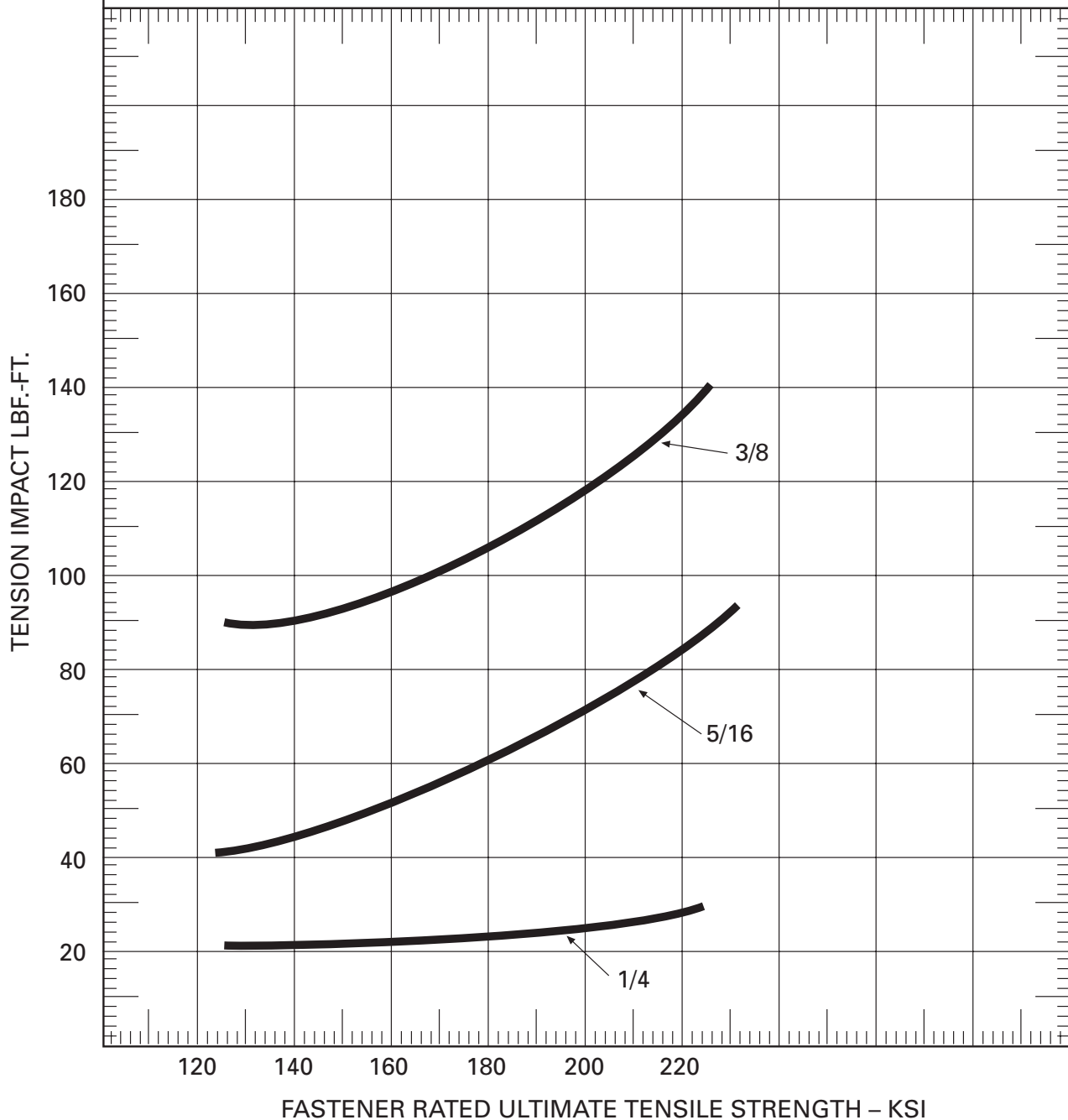


FIG. 22

PRODUCT ENGINEERING BULLETIN

UNBRAKO PRODUCT ENGINEERING BULLETIN

Standard Inch Socket Head Cap Screws Are Not Grade 8 Fasteners

There is a common, yet reasonable, misconception that standard, inch, alloy steel socket head cap screws are "Grade 8". This is not true. The misconception is reasonable because "Grade 8" is a term generally associated with "high strength" fasteners. A person desiring a "high strength" SHCS may request a "Grade 8 SHCS". This is technically incorrect for standard SHCSs. The term Grade 8 defines specific fastener characteristics which must

be met to be called "Grade 8". Three of the most important characteristics are not consistent with requirements for industry standard SHCSs: tensile strength, hardness, and head marking. Some basic differences between several fastener classifications are listed below. The list is not comprehensive but intended to provide a general understanding. SHCSs can be manufactured to meet Grade 8 requirements on a special order basis.

Fastener Designation	Grade 2	Grade 5	Grade 8	Industry SHCS	Unbrako SHCS
Applicable Standard	SAE J429	SAE J429	SAE J429	ASTM A574	ASTM A574 SPS-B-271
Strength Level, UTS KSI, min.	74 (1/4-3/4) 60 (7/8 - 1 1/2)	120 (1/4 - 1) 105 (1 1/8 - 1 1/2)	150 (1/4 - 11/2)	180 (≤1/2) 170 (> 1/2)	190 (≤ 1/2) 180 (> 1/2)
Hardness, Rockwell	B80-B100 B70-B100	C25-C34 C19-C30	C33-C39	C39-C45 C37-C45	C39-C43 C38-C43
General Material Type	Low or Medium Carbon Steel	Medium Carbon Steel	Medium Carbon Alloy Steel	Medium Carbon Alloy Steel	Medium Carbon Alloy Steel
Identification Requirement	None	Three Radial Lines	Six Radial Lines	SHCS Configuration	Mfr's ID
Typical Fasteners	Bolts Screws Studs Hex Heads	Bolts Screws Studs Hex Heads	Bolts Screws Studs Hex Heads	Socket Head Cap Screws	Socket Head Cap Screws

THREADS IN BOTH SYSTEMS

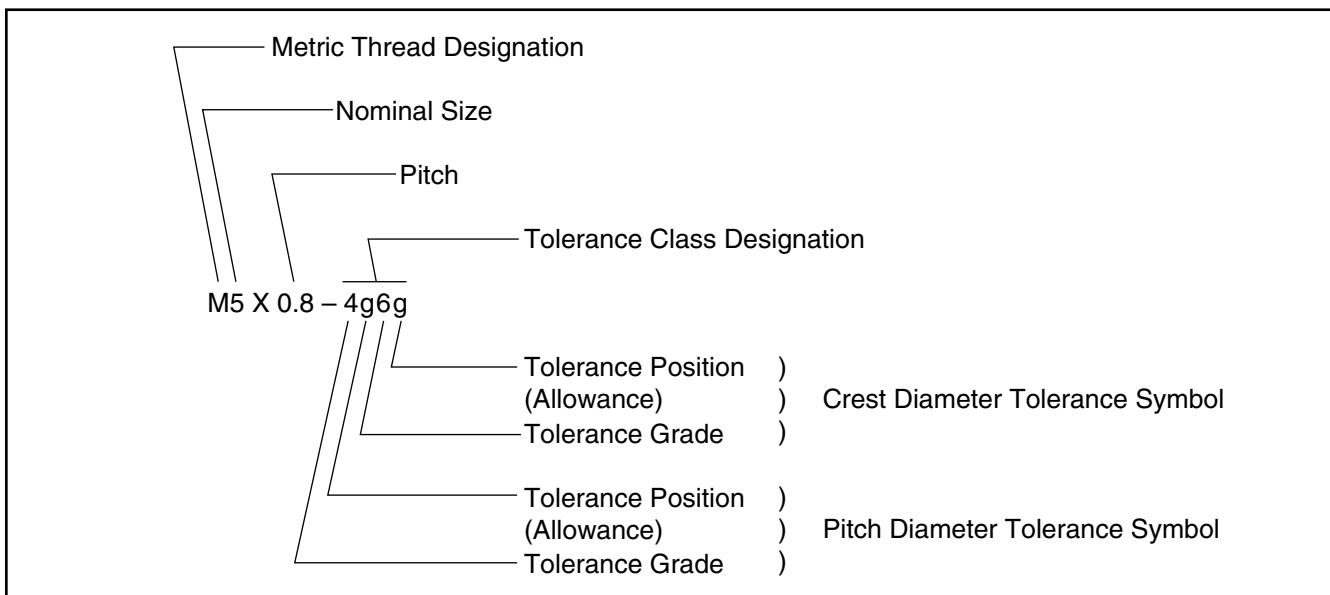
Thread forms and designations have been the subject of many long and arduous battles through the years. Standardization in the inch series has come through many channels, but the present unified thread form could be considered to be the standard for many threaded products, particularly high strength ones such as socket head cap screws, etc. In common usage in U.S.A., Canada and United Kingdom are the Unified National Radius Coarse series, designated UNRC, Unified National Radius Fine series, designated UNRF, and several special series of various types, designated UNS. This thread, UNRC or UNRF, is designated by specifying the diameter and threads per inch along with the suffix indicating the thread series, such as 1/4 - 28 UNRF. For threads in Metric units, a similar approach is used, but with some slight variations. A diameter and pitch are used to designate the series, as in the Inch system, with modifications as follows: For coarse threads, only the prefix M and the diameter are necessary, but for fine threads, the pitch is shown as a suffix. For example, M16 is a coarse thread designation representing a diameter of 16 mm with a pitch of 2 mm understood. A similar fine thread part would be M16 x 1.5 or 16 mm diameter with a pitch of 1.5 mm.

For someone who has been using the Inch system, there are a couple of differences that can be a little confusing. In the Inch series, while we refer to threads per inch as pitch; actually the number of threads is 1/pitch. Fine threads are referenced by a larger number than coarse threads because they "fit" more threads per inch.

In Metric series, the diameters are in millimeters, but the pitch is really the pitch. Consequently the coarse thread has the large number. The most common metric thread is the coarse thread and falls generally between the inch coarse and fine series for a comparable diameter.

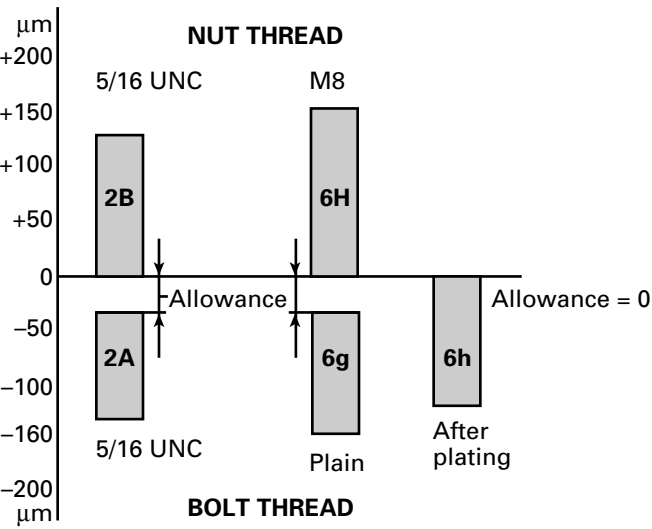
Also to be considered in defining threads is the tolerance and class of fit to which they are made. The International Standards Organization (ISO) metric system provides for this designation by adding letters and numbers in a certain sequence to the callout. For instance, a thread designated as M5 x 0.8 4g6g would define a thread of 5 mm diameter, 0.8 mm pitch, with a pitch diameter tolerance grade 6 and allowance "g". These tolerances and fields are defined as shown below, similar to the Federal Standard H28 handbook, which defines all of the dimensions and tolerances for a thread in the inch series. The callout above is similar to a designation class 3A fit, and has a like connotation.

COMPLETE DESIGNATIONS



METRIC THREADS

Example of thread tolerance positions and magnitudes.
Comparison 5/16 UNC and M8. Medium tolerance grades – Pitch diameter.



DEVIATIONS

external	internal	basic clearance
h g e	H G	none small large

NOTE:
Lower case letters = external threads
Capital letters = internal threads

THROUGH-HOLE PREPARATION

DRILL AND COUNTERBORE SIZES FOR INCH SOCKET HEAD CAP SCREWS

Note 1

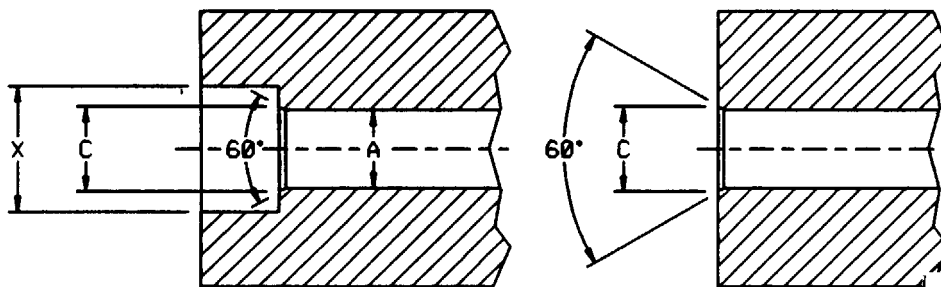
Close Fit: Normally limited to holes for those lengths of screws threaded to the head in assemblies in which: (1) only one screw is used; or (2) two or more screws are used and the mating holes are produced at assembly or by matched and coordinated tooling.

Note 2

Normal Fit: Intended for: (1) screws of relatively long length; or (2) assemblies that involve two or more screws and where the mating holes are produced by conventional tolerancing methods. It provides for the maximum allowable eccentricity of the longest standard screws and for certain deviations in the parts being fastened, such as deviations in hole straightness; angularity between the axis of the tapped hole and that of the hole for the shank; differences in center distances of the mating holes and other deviations.

Note 3

Chamfering: It is considered good practice to chamfer or break the edges of holes that are smaller than "F" maximum in parts in which hardness approaches, equals or exceeds the screw hardness. If holes are not chamfered, the heads may not seat properly or the sharp edges may deform the fillets on the screws, making them susceptible to fatigue in applications that involve dynamic loading. The chamfers, however, should not be larger than needed to ensure that the heads seat properly or that the fillet on the screw is not deformed. Normally, the chamfers do not need to exceed "F" maximum. Chamfers exceeding these values reduce the effective bearing area and introduce the possibility of indentation when the parts fastened are softer than screws, or the possibility of brinelling of the heads of the screws when the parts are harder than the screws. (See "F" page 6).



nominal size	basic screw diameter	A				X	C	hole dimensions			
		drill size for hole A				counter- bore diameter	countersink diameter D Max. + 2F(Max.)	tap drill size		**body drill size	counter- bore size
		close fit		normal fit							
		nom.	dec.	nom.	dec.			UNRC	UNRF		
0	0.0600	51*	0.0670	49*	0.0730	1/8	0.074	–	3/64	#51	1/8
1	0.0730	46*	0.0810	43*	0.0890	5/32	0.087	1.5mm	#53	#46	5/32
2	0.0860	3/32	0.0937	36*	0.1065	3/16	0.102	#50	#50	3/32	3/16
3	0.0990	36*	0.1065	31*	0.1200	7/32	0.115	#47	#45	#36	7/32
4	0.1120	1/8	0.1250	29*	0.1360	7/32	0.130	#43	#42	1/8	7/32
5	0.1250	9/64	0.1406	23*	0.1540	1/4	0.145	#38	#38	9/64	1/4
6	0.1380	23*	0.1540	18*	0.1695	9/32	0.158	#36	#33	#23	9/32
8	0.1640	15*	0.1800	10	0.1935	5/16	0.188	#29	#29	#15	5/16
10	0.1900	5*	0.2055	2*	0.2210	3/8	0.218	#25	#21	#5	3/8
1/4	0.2500	17/64	0.2656	9/23	0.2812	7/16	0.278	#7	#3	17/64	7/16
5/16	0.3125	21/64	0.3281	11/32	0.3437	17/32	0.346	F	I	21/64	17/32
3/8	0.3750	25/64	0.3906	13/32	0.4062	5/8	0.415	5/16	Q	25/64	5/8
7/16	0.4375	29/64	0.4531	15/32	0.4687	23/32	0.483	U	25/64	29/64	23/32
1/2	0.5000	33/64	0.5156	17/32	0.5312	13/16	0.552	27/64	29/64	33/64	13/16
5/8	0.6250	41/64	0.6406	21/32	0.6562	1	0.689	35/64	14.5mm	41/64	1
3/4	0.7500	49/64	0.7656	25/32	0.7812	1-3/16	0.828	21/32	11/16	49/64	1-3/16
7/8	0.8750	57/64	0.8906	29/32	0.9062	1-3/8	0.963	49/64	20.5mm	57/64	1-3/8
1	1.0000	1-1/64	1.0156	1-1/32	1.0312	1-5/8	1.100	7/8	59/64	1-1/64	1-5/8
1-1/4	1.2500	1-9/32	1.2812	1-5/16	1.3125	2	1.370	1-7/64	1-11/64	1-9/32	2
1-1/2	1.5000	1-17/32	1.5312	1-9/16	1.5625	2-3/8	1.640	34mm	36mm	1-17/32	2-3/8

** Break edge of body drill hole to clear screw fillet.

DRILL AND COUNTERBORE SIZES

DRILL AND COUNTERBORE SIZES FOR METRIC SOCKET HEAD CAP SCREWS

Note 1

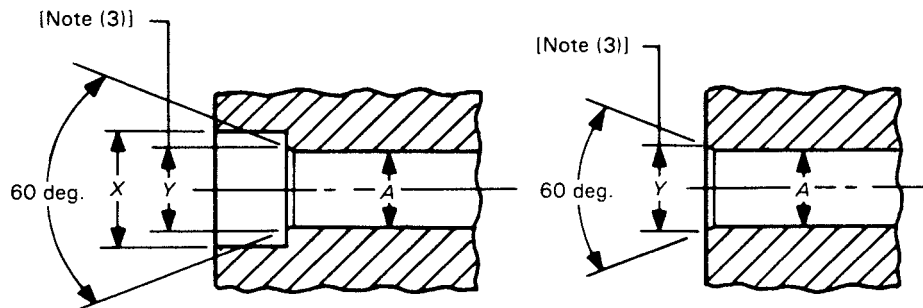
Close Fit: Normally limited to holes for those lengths of screws threaded to the head in assemblies in which: (1) only one screw is used; or (2) two or more screws are used and the mating holes are produced at assembly or by matched and coordinated tooling.

Note 2

Normal Fit: Intended for: (1) screws of relatively long length; or (2) assemblies that involve two or more screws and where the mating holes are produced by conventional tolerancing methods. It provides for the maximum allowable eccentricity of the longest standard screws and for certain deviations in the parts being fastened, such as deviations in hole straightness; angularity between the axis of the tapped hole and that of the hole for the shank; differences in center distances of the mating holes and other deviations.

Note 3

Chamfering: It is considered good practice to chamfer or break the edges of holes that are smaller than "B" maximum in parts in which hardness approaches, equals or exceeds the screw hardness. If holes are not chamfered, the heads may not seat properly or the sharp edges may deform the fillets on the screws, making them susceptible to fatigue in applications that involve dynamic loading. The chamfers, however, should not be larger than needed to ensure that the heads seat properly or that the fillet on the screw is not deformed. Normally, the chamfers do not need to exceed "B" maximum. Chamfers exceeding these values reduce the effective bearing area and introduce the possibility of indentation when the parts fastened are softer than screws, or the possibility of brinelling of the heads of the screws when the parts are harder than the screws.



Nominal Size or Basic Screw Diameter	A		X	Y	B
	Nominal Drill Size		Counterbore Diameter	Countersink Diameter [Note (3)]	Transition Diameter, Max.
	Close Fit [Note (1)]	Normal Fit [Note (2)]			
M1.6	1.80	1.95	3.50	2.0	2.0
M2	2.20	2.40	4.40	2.6	2.6
M2.5	2.70	3.00	5.40	3.1	3.1
M3	3.40	3.70	6.50	3.6	3.6
M4	4.40	4.80	8.25	4.7	4.7
M5	5.40	5.80	9.75	5.7	5.7
M6	6.40	6.80	11.25	6.8	6.8
M8	8.40	8.80	14.25	9.2	9.2
M10	10.50	10.80	17.25	11.2	11.2
M12	12.50	12.80	19.25	14.2	14.2
M14	14.50	14.75	22.25	16.2	16.2
M16	16.50	16.75	25.50	18.2	18.2
M20	20.50	20.75	31.50	22.4	22.4
M24	24.50	24.75	37.50	26.4	26.4
M30	30.75	31.75	47.50	33.4	33.4
M36	37.00	37.50	56.50	39.4	39.4
M42	43.00	44.0	66.00	45.6	45.6
M48	49.00	50.00	75.00	52.6	52.6

HARDNESS – TENSILE CONVERSION

INCH ROCKWELL – BRINELL – TENSILE CONVERSION

Rockwell "C" scale	Brinell hardness number	tensile strength approx. 1000 psi
60	654	336
59	634	328
58	615	319
57	595	310
56	577	301
55	560	292
54	543	283
53	524	274
52	512	265
51	500	257
50	488	249
49	476	241
48	464	233
47	453	225
46	442	219
45	430	212
44	419	206

Rockwell "C" scale	Brinell hardness number	tensile strength approx. 1000 psi
43	408	200
42	398	194
41	387	188
40	377	181
39	367	176
38	357	170
37	347	165
36	337	160
35	327	155
34	318	150
33	309	147
32	301	142
31	294	139
30	285	136
29	279	132
28	272	129
27	265	126

Rockwell		Brinell hardness number	tensile strength approx. 1000 psi
"C" scale	"B" scale		
26		259	123
25		253	120
24		247	118
23		241	115
22	100	235	112
21	99	230	110
20	98	225	107
(19)		220	104
(18)	97	215	103
(17)		210	102
(16)	96	206	100
(15)		201	99
(14)	95	197	97
(13)	94	193	96
(12)	93	190	93
(11)		186	91
(10)	92	183	90

METRIC ROCKWELL – BRINELL – TENSILE CONVERSION

Rockwell "C" scale	Brinell hardness number	tensile strength approx. MPa
60	654	2,317
59	634	2,261
58	615	2,199
57	595	2,137
56	577	2,075
55	560	2,013
54	543	1,951
53	524	1,889
52	512	1,827
51	500	1,772
50	488	1,717
49	476	1,662
48	464	1,606
47	453	1,551
46	442	1,510
45	430	1,462
44	419	1,420

Rockwell "C" scale	Brinell hardness number	tensile strength approx. MPa
43	408	1,379
42	398	1,338
41	387	1,296
40	377	1,248
39	367	1,213
38	357	1,172
37	347	1,138
36	337	1,103
35	327	1,069
34	318	1,034
33	309	1,014
32	301	979
31	294	958
30	285	938
29	279	910
28	272	889
27	265	869

Rockwell		Brinell hardness number	tensile strength approx. MPa
"C" scale	"B" scale		
26		259	848
25		253	827
24		247	814
23		241	793
22	100	235	772
21	99	230	758
20	98	225	738
(19)		220	717
(18)	97	215	710
(17)		210	703
(16)	96	206	690
(15)		201	683
(14)	95	197	669
(13)	94	193	662
(12)	93	190	641
(11)		186	627
(10)	92	183	621

THREAD STRESS AREAS

Inch and Metric

STRESS AREAS FOR THREADED FASTENERS – INCH

Diameter (in.)		Diameter (mm)	Threads Per in.		Square Inches		
					Tensile Stress Area Per H-28		Nominal Shank
			UNRC	UNRF	UNRC	UNRF	
#0	0.06	1.52	–	80	–	0.00180	0.002827
#1	0.07	1.85	64	72	0.00263	0.00278	0.004185
#2	0.09	2.18	56	64	0.00370	0.00394	0.005809
#3	0.10	2.51	48	56	0.00487	0.00523	0.007698
#4	0.11	2.84	40	48	0.00604	0.00661	0.009852
#5	0.13	3.18	40	44	0.00796	0.00830	0.012272
#6	0.14	3.51	32	40	0.00909	0.01015	0.014957
#8	0.16	4.17	32	36	0.0140	0.01474	0.021124
#10	0.19	4.83	24	32	0.0175	0.0200	0.028353
1/4	0.25	6.35	20	28	0.0318	0.0364	0.049087
5/16	0.31	7.94	18	24	0.0524	0.0580	0.076699
3/8	0.38	9.53	16	24	0.0775	0.0878	0.11045
7/16	0.44	11.11	14	20	0.1063	0.1187	0.15033
1/2	0.50	12.70	13	20	0.1419	0.1599	0.19635
9/16	0.56	14.29	12	18	0.182	0.203	0.25
5/8	0.63	15.88	11	18	0.226	0.256	0.31
3/4	0.75	19.05	10	16	0.334	0.373	0.44179
7/8	0.88	22.23	9	14	0.462	0.509	0.60132
1	1.00	25.40	8	12	0.606	0.663	0.79
1-1/8	1.13	28.58	7	12	0.763	0.856	0.99402
1-1/4	1.25	31.75	7	12	0.969	1.073	1.2272
1-3/8	1.38	34.93	6	12	1.155	1.315	1.4849
1-1/2	1.50	38.10	6	12	1.405	1.581	1.7671
1-3/4	1.75	44.45	5	12	1.90	2.19	2.4053
2	2.00	50.80	4-1/2	12	2.50	2.89	3.1416
2-1/4	2.25	57.15	4-1/2	12	3.25	3.69	3.9761
2-1/2	2.50	63.50	4	12	4.00	4.60	4.9088
2-3/4	2.75	69.85	4	12	4.93	5.59	5.9396
3	3.00	76.20	4	12	5.97	6.69	7.0686

STRESS AREAS FOR THREADED FASTENERS – METRIC

Nominal Dia. Thread and Pitch (mm)	Thread Tensile Stress Area (mm ²)	Nominal Shank Area (mm ²)
1.6 x 0.35	1.27	2.01
2.0 x 0.4	2.07	3.14
2.5 x 0.45	3.39	4.91
3.0 x 0.5	5.03	7.07
4.0 x 0.7	8.78	12.6
5.0 x 0.8	14.2	19.6
6.0 x 1	20.1	28.3
8.0 x 1.25	36.6	50.3
10 x 1.5	58.00	78.5
12 x 1.75	84.3	113
14 x 2	115	154
16 x 2	157	201

Nominal Dia. Thread and Pitch (mm)	Thread Tensile Stress Area (mm ²)	Nominal Shank Area (mm ²)
18 x 2.5	192	254
20 x 2.5	245	314
22 x 2.5	303	380
24 x 3	353	452
27 x 3	459	573
30 x 3.5	561	707
33 x 3.5	694	855
36 x 4	817	1018
42 x 4.5	1120	1385
48 x 5	1470	1810

ENGINEERING PART NUMBERS – INCH

Unbrako provides a stock number for every standard, stocked item in its price list. However, there may be particular sizes or optional features the user may desire. The following part numbering system allows the engineer or designer to record a particular description for ordering.

Alloy Steel	Drilled Head (3)	#4	UNRC	1 1/2"	Cadmium Plate
20097	H3	-94	C	-24	C

FINISH

B – Chemical Black Oxide S – Silver Plate
C – Cadmium Plate – Silver U – Zinc Plate – Silver
D – Cadmium Plate – Yellow Z – Zinc Plate – Yellow

No letter indicates standard black finish (Thermal Oxide) for alloy steel and passivation for stainless steel.

LENGTH in 16ths

THREAD TYPE C – coarse, F – fine

DIAMETER*

DIA.	#0	#1	#2	#3	#4	#5	#6	#8	#10	1/4	5/16	3/8	7/16	1/2	9/16
DASH NO.	90	91	92	93	94	95	96	98	3	4	5	6	7	8	9

DIA.	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8	1 1/2	1 3/4	2	2 1/4	2 1/2	2 3/4	3
DASH NO.	10	12	14	16	18	20	22	24	28	32	36	40	44	48

OPTIONAL FEATURES

Cross Drilled Heads: Self-Locking:
H1 – 1 Hole Thru E – LOC-WEL to MIL-DTL18240
H2 – 2 Hole2 Thru L – LOC-WEL (Commercial)
H3 – 3 Holes Thru P – Nylon Plug
TF – TRU-FLEX
K – Nylon Plug to MIL-DTL18240

BASE NUMBER

20097 – socket head cap screw – alloy steel
20098 – socket head cap screw – stainless steel
72531 – low head cap screw
12705 – shoulder screw
16990 – flat head cap screw – alloy steel
16991 – flat head cap screw – stainless steel
38030 – button head cap screw – alloy
38031 – button head cap screw – stainless steel
05455 – square head cap screw – knurled cup
05456 – square head cap screw – half dog

Set Screws

Alloy Steel	Stainless Steel	
28700	28707	flat point
28701	28708	cup point
28704	28709	knurled cup point
28702	28710	cone point
28705	28711	oval point
28706	28713	half dog point

* Shoulder screws are designated by shoulder diameter

OPTIONAL PART NUMBERING SYSTEM

PRESSURE PLUG PART NUMBERS

Basic Part No.	Material	1/4"	Finish	FINISH
29466	A	-4	C	

B – Chemical Black Oxide
 C – Cadmium Plate-Silver
 D – Cadmium Plate-Yellow
 S – Silver Plate
 U – Zinc Plate – Silver
 Z – Zinc Plate – Yellow

A – Austenitic Stainless
 D – Aluminum
 E – Brass
 No letter – alloy steel

NOMINAL SIZE IN 16ths
 OPTIONAL FEATURES
 BASIC PART NUMBER

* Standard stock available in austenitic stainless steel, brass, and alloy only
 ** Standard stock available in austenitic stainless steel, and alloy only

29466 – dry seal
 *38194 – LEVEL-SEAL
 **69188 – PTFE/TEFLON coated

DOWEL PINS PART NUMBERS

dowel pin	1/4"	.001 oversize	1/2"	
28420	-250	B	-8	

The Part number consists of (1) a basic part number describing the item; (2) a dash number and letter designating diameter and oversize dimension; (3) a dash number designating length.

LENGTH in 16ths
 OVERSIZE A-.0002, B-.001, C-.002 (see below)
 DIAMETER in thousandths
 BASIC PART NUMBER

28420 – Standard Dowel Pins
 69382 – Pull-Out Dowel Pins

HEX KEYS PART NUMBERS

long arm	1/4"	
05854	-13	

The Part number consists of (1) a basic part number describing the item; (2) a dash number designating size and a letter denoting finish.

FINISH Standard Black Finish (Thermal Oxide)

See dash number in dimension table page 32
 BASIC PART NUMBER

05853 – short arm wrench
 05854 – long arm wrench
 78950-6" – long arm wrench

ENGINEERING PART NUMBERS – METRIC

Alloy Steel	Drilled Head (3)	4MM Dia.	Thread Pitch	Length	Cadmium Plate
76000	H3	-M4	-0.7	-12	C

FINISH

B – Chemical Black Oxide S – Silver Plate
 C – Cadmium Plate – Silver U – Zinc Plate – Silver
 D – Cadmium Plate – Yellow Z – Zinc Plate – Yellow

No letter indicates standard black finish (Thermal Oxide) for alloy steel and passivation for stainless steel.

LENGTH in mm

THREAD TYPE STATE THREAD PITCH

DIAMETER in mm*

OPTIONAL FEATURES

Cross Drilled Heads: Self-Locking:
 H1 – 1 Hole Thru E – LOC-WEL to MIL-DTL-18240
 H2 – 2 Hole2 Thru L – LOC-WEL (Commercial)
 H3 – 3 Holes Thru P – Nylon Plug
 TF – TRU-FLEX
 K – Nylon Plug to MIL-DTL-18240

BASE NUMBER

76000 – metric socket head cap screw – alloy steel
 76001 – metric socket head cap screw – stainless steel
 76002 – metric low head cap screw – alloy
 76032 – metric low head cap screw – stainless steel
 76005 – metric flat head cap screw – alloy steel
 76006 – metric flat head cap screw – stainless steel
 76003 – metric button head cap screw – alloy
 76004 – metric button head cap screw – stainless steel
 76007 – metric shoulder screw – alloy

Metric Set Screws

Alloy Steel	Stainless Steel	
76010	76016	flat point
76011	76017	cup point
76012	76018	knurled cup point
76013	76019	cone point
76014	76020	oval point
76015	76021	half dog point

* Shoulder screws are designated by shoulder diameter

HEX KEYS PART NUMBERS (METRIC)

The Part number consists of (1) a basic part number describing the item; (2) a dash number designating size.

long arm	5mm
76023	5

FINISH Standard Black Finish (Thermal Oxide)

Key size in mm

BASIC PART NUMBER | 76022 – short arm wrench
76023 – long arm wrench

DOWEL PINS PART NUMBERS (METRIC)

The Part number consists of (1) a basic part number describing the item; (2) a dash number and letter designating diameter and oversize dimension; (3) a dash number designating length.

dowel pin	6mm	.0275 oversize	8mm
76024	-6	B	-8

LENGTH in mm

OVERSIZE A-.0055, B-.0275mm

DIAMETER in mm

BASIC PART NUMBER | 76024 – Standard Dowel Pins
76035 – Pull-Out Dowel Pins



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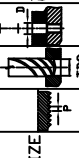
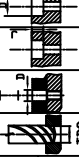
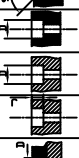
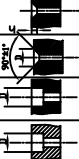
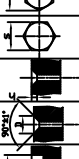
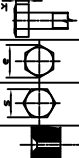
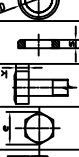
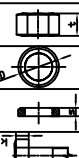
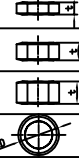
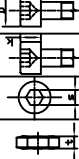
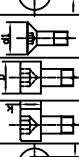
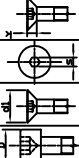
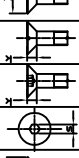
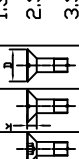
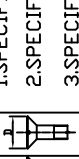
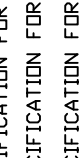
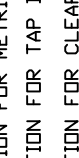
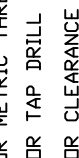
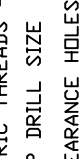
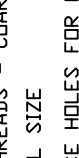
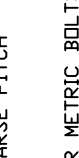
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THREAD CONVERSION CHARTS

DIAMETER/THREAD PITCH COMPARISON

INCH SERIES			METRIC			
Size	Dia. (In.)	TPI	Size	Dia.(In.)	Pitch (mm)	TPI (Approx)
#0	0.060	80				
			M1.6	0.063	0.35	74
#1	0.073	64				
			M2	0.079	0.4	64
#2	0.096	56				
			M2.5	0.098	0.45	56
#3	0.099	48				
#4	0.112	40				
			M3	0.118	0.5	51
#5	0.125	40				
#6	0.138	32				
			M4	0.157	0.7	36
#8	0.164	32				
#10	0.190	24				
			M5	0.196	0.8	32
			M6	0.236	1.00	25
1/4	0.250	20				
5/16	0.312	18				
			M8	0.315	1.25	20
3/8	0.375	16				
			M10	0.393	1.5	17
7/16	0.437	14				
			M12	0.472	1.75	14.5
1/2	0.500	13				
			M14	0.551	2	12.5
5/8	0.625	11				
			M16	0.63	2	12.5
3/4	0.750	10				
			M20	0.787	2.5	10
7/8	0.875	9				
			M24	0.945	3	8.5
1	1.000	8				
			M27	1.063	3	8.5

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
SIZE																								
M3	0.5	2.5	3.4	3.4	6.3	2.4	—	1.7	5.5	6.4	2.0	0.5	7	—	2.4	1.6	2.5	3	5.5	6	2	1.7	1.0	5
M4	0.7	3.3	4.5	4.6	8.0	3.2	9.0	2.3	7.0	8.1	2.8	0.8	9	—	3.2	2.0	3	4	7	8	2.5	2.3	1.5	7
M5	0.8	4.2	5.5	5.7	9.5	4.0	11.0	2.8	9	10.4	3.5	1.0	10	—	4.0	2.5	4	5	8.5	10	3	2.8	2.0	9
M6	1.0	5.0	6.6	6.8	11.0	4.7	13.0	3.3	10	11.5	4.5	1.6	12.5	—	5.0	3	5	6	10	12	4	3.3	2.0	10
M8	1.25	6.8	9	9.0	14.5	6.0	17.2	4.4	13	16.2	5.5	1.6	17	8	6.5	4	6	8	13	16	5	4.4	—	—
M10	1.5	8.5	11	11.0	18.0	7.0	21.5	5.5	17	19.6	7	2.0	21	10	8.0	5	8	10	16	20	6	5.5	—	—
M12	1.75	10.25	14	13.0	20	8.0	26.0	6.5	19	21.8	8	2.5	24	12	10	7	10	12	18	24	8	6.5	—	—
M14	2.0	12	16	15.0	24	8.3	28.8	7.5	22	25.4	9	2.5	28	14	11	8	—	14	21	—	—	—	—	—
M16	2.0	14	18	16.5	26	10.5	34.0	8.5	24	27.7	10.5	3	30	16	13	8	14	16	24	30	10	7.5	—	—
M18	2.5	15.5	20	19.5	29	11.5	34.8	9.5	27	31.2	13	3	34	18	15	9	—	18	27	—	—	—	—	—
M20	2.5	17.5	22	21.5	33	12.5	42.0	10.5	30	34.6	13	3	37	20	16	9	17	20	30	36	12	8.5	—	—
M22	2.5	19.5	24	23.5	36	13.5	—	—	32	36.9	15	3	40	22	17	—	—	22	33	—	—	—	—	—
M24	3.0	21	26	25.5	39	13.5	—	—	36	41.6	15	4	44	24	18	—	19	24	36	39	14	14	—	—
M27	3.0	24	30	28.5	42	16.5	—	—	44	47.3	19	4	50	27	20	—	—	—	—	—	—	—	—	—
M30	3.5	26.5	33	32.0	48	17.5	—	—	46	53.1	19	4	56	30	22	—	22	—	45	—	—	—	—	—
M33	3.5	29.5	36	—	—	—	—	—	50	57.7	—	5	61	33	25	—	—	—	—	—	—	—	—	—
M36	4.0	32	39	38	57	—	—	—	55	63.5	23	5	66	36	28	—	27	—	54	—	—	—	—	—
M39	4.0	35	42	—	—	—	—	—	60	—	—	6	75	39	30	—	—	—	—	—	—	—	—	—
IS NO	IS 4218 - 1967	IS 4218 - 1967	IS 1369 - 1975	IS 3406 PART 2 -1975	IS 2269 - 1967	IS 1366 - 1968	IS 3406 PART 2 -1975	IS 3406 PART 2 -1975	IS 1363 - 1967	IS 1363 - 1967	IS 1363 - 1967	IS 2269 - 1967	IS 2269 - 1967		IS 1363&1364-1967 # IS 2389-1968	IS 1363&1364-1967 # IS 2389-1968	IS 2269 - 1967	IS 2269 - 1967	IS 2269 - 1967	IS 6761 - 1972	IS 6761 - 1972	IS 5308 - 1969	IS 5308 - 1969	IS 5308 - 1969

*** MUST BE OF SECONDARY PREFERENCE**

- 1.SPECIFICATION FOR METRIC THREADS – COARSE PITCH
- 2.SPECIFICATION FOR TAP DRILL SIZE
- 3.SPECIFICATION FOR CLEARANCE HOLES FOR METRIC BOLTS
- 4.SPECIFICATION FOR COUNTER BORES (DEPTH) – SHCS
- 5.SPECIFICATION FOR COUNTER BORES (DIAMETER) – SHCS
- 6.SPECIFICATION FOR COUNTER BORES (DIAMETER) – SLOTTED CHEESE HEAD SCREWS
- 7.SPECIFICATION FOR COUNTER SINKS (DIAMETER) – SLOTTED HEAD COUNTERSUNK SCREWS
- 8.SPECIFICATION FOR COUNTER SINKS (DEPTH) – SLOTTED HEAD COUNTERSUNK SCREWS
- 9.SPECIFICATION HEXAGONAL BOLTS – ACROSS FLAT
- 10.SPECIFICATION HEXAGONAL BOLTS – ACROSS CORNER
- 11.SPECIFICATION HEXAGONAL BOLTS – HEAD THICKNESS
- 12.SPECIFICATION FOR MACHINED WASHERS – THICKNESS
- 13.SPECIFICATION FOR MACHINED WASHERS – OUTER DIAMETER
- 14.SPECIFICATION FOR HEXAGONAL NUTS (THICK) – THICKNESS
- 15.SPECIFICATION FOR HEXAGONAL NUTS (MEDIUM) – THICKNESS
- 16.SPECIFICATION FOR HEXAGONAL NUTS (LOCK NUT) – THICKNESS
- 17.SPECIFICATION FOR HEXAGONAL SOCKET HEAD CAP SCREWS – WIDTH ACROSS FLAT
- 18.SPECIFICATION FOR HEXAGONAL SOCKET HEAD CAP SCREWS – HEAD THICKNESS
- 19.SPECIFICATION FOR HEXAGONAL SOCKET HEAD CAP SCREWS – HEAD DIAMETER
- 20.SPECIFICATION FOR COUNTERSUNK HEAD SCREWS WITH HEXAGONAL SOCKET- HEAD DIAMETER
- 21.SPECIFICATION FOR COUNTERSUNK HEAD SCREWS WITH HEXAGONAL SOCKET- WIDTH ACROSS FLAT
- 22.SPECIFICATION FOR COUNTERSUNK HEAD SCREWS WITH HEXAGONAL SOCKET- HEAD THICKNESS
- 23.SPECIFICATION FOR SLOTTED COUNTERSUNK HEAD SCREWS – HEAD THICKNESS
- 24.SPECIFICATION FOR SLOTTED COUNTERSUNK HEAD SCREWS – HEAD DIAMETER

PREPARED BY:
RADHANADEFCWADAN